STEAM CHARTS

ALSO

A Table of Theoretical Jet Velocities *and* The Corrections of Mercury Columns

WITH

FIFTY ILLUSTRATIVE PROBLEMS

BY

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PREFACE

THIS little book is intended to be of assistance to engineers and students when making calculations involving wet or superheated steam. The chief aim of the author has been to prepare a set of steam charts which shall be accurate and comprehensive, and at the same time convenient to handle, and easy to read. An attempt has also been made to give, concisely, the corrections to be applied to the readings of mercury columns, and to prepare a table of velocities, which it is hoped may prove useful.

In order to illustrate some of the uses of the charts and tables, and also to aid those who may desire it, a number of problems, with their solutions, have been added. To make these of more assistance, they have been indexed. For the further aid of those who may desire a brief review of the thermodynamics of steam, and in order to make clear the meaning of all terms used, the few pages of Fundamental Principles were written.

For the main chart, total heat and specific volume were chosen as coordinates because of the fact that upon these two values could be plotted lines of constant pressure, entropy and quality (or superheat), so that each pair of the five sets of lines will make clear intersections. The total heat entropy chart does not permit this. To complete the set of values ordinarily needed, the curve was added, showing the heat of the liquid and temperature of vaporization. The supplementary chart, Plate 8, enables one to read the external work, and therefore obtain easily the intrinsic heat. The index chart for Plates 1 to 6 was made to give a general idea of the relative position and shape of each set of lines, to show quickly the limiting values for each of the six sections, and to assist in determining the particular plate needed.

The range of pressures, qualities, and superheats is intended to be more than sufficient for present practice. For the wet region the inch of mercury was used as the main unit to represent pressures

less than one pound absolute, as it is believed that this is the more convenient one for practical work. Special endeavor has been made to prevent confusion of these two units by using broken lines to represent pressures in inches of mercury, and by putting the proper units with each numeral representing pressure in this region.

The book form of chart was chosen because the author believes that it will be of greater convenience and easier to read than a large folded chart made to the same scales. By making the plates small the eye has to travel only a short distance to read the scales, and this may also be done without requiring any desk space whatever. The book form also has the advantages of better protecting the chart, permitting a quicker reference, and wasting less space in the corners, than does the same chart when in the form of a large folded sheet.

To Prof. Lionel S. Marks and to Dr. Harvey N. Davis, and to their publishers, Longmans, Green & Co., the author desires to express his thanks for permission to use their steam tables in preparing these charts. He also wishes to acknowledge his indebtedness to Prof. Albert W. Smith, Director of Sibley College, and to Prof. William N. Barnard, for their many helpful criticisms; and to Mr. C. H. Berry and Mr. E. T. Jones, instructors in Sibley College, for their able assistance in preparing the charts and problems.

F. O. E.

ITHACA, New York, August, 1914.

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INTRODUCTION

FUNDAMENTAL PRINCIPLES

Pressure-Volume and Temperature-Entropy Diagrams.—In the study of thermodynamics of vapors the pressure-volume and the temperature-entropy diagrams are of very great importance, for the reason that by their aid the areas representing work and heat, respectively, may usually be shown. Since the engineer is concerned with the transformation of heat into work, any diagrams which will assist him to understand how this is accomplished will always be very useful. Other diagrams may be of more assistance in obtaining numerical results, but these two will always be foremost in analyzing thermodynamic processes.

Pressure-Volume Diagram.—Referring to Fig. 1a, where the absolute pressure, P, in pounds per square foot, is represented by the ordinates, and the volume, V, in cubic feet by abscissæ, the area *abhg* underneath curve *ab* represents the work done in foot-pounds by the substance in passing along the constant-pressure line from *a* to *b*. This might be expressed in this manner:

$$\text{Work}\Big]_a^b = \int_{V_a}^{V_b} PdV = P\,[V_b - V_a] = \text{area } abhg.$$

In general, it may be stated that if a substance expands or is compressed in such a manner that its pressure-volume history is definite for the entire change, as from the point *a* to some point *c* along the path *adc*, Fig. 1a, the following equation is true:

$$\text{Work}\Big]_a^c = \int_{V_a}^{V_c} PdV = \text{area } adchg.$$

This would represent work done by the substance upon some other body. In case the substance had been compressed along the

path *cda*, then the area *cdagh* represents the work done upon this substance in order to compress it,

$$\text{or Work}\Big]_c^a = \int_{V_c}^{V_a} PdV = -\int_{V_a}^{V_c} PdV$$
$$= -\,[\text{area } adchg] = \text{area } cdagh.$$

Temperature-Entropy Diagram.—If, by the addition of heat, a substance may be made to change its state point *a* to some other

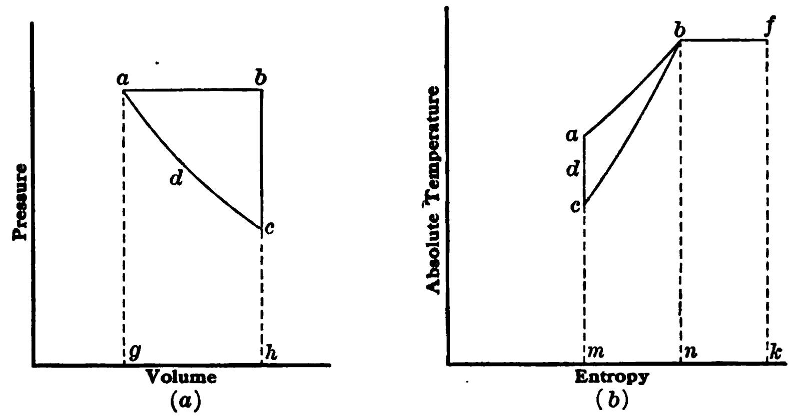


FIG. 1.

condition *b*, in such a manner that its temperature and pressure are uniform* throughout the entire mass of the substance during this change, then the heat which has been added to this substance is equal to the area *mabn*, Fig. 1b. This is true by reason of the definition of entropy. In other words, for *such processes as this*, if

H = Heat added to the substance in B. t. u.
T = Absolute temperature (= 460 + t° F.)
ϕ = Entropy

then the change of entropy from *a* to *b* is defined by the equation

$$d\phi = \frac{dH}{T}$$

$$\text{or H}\Big]_a^b = \int_{\phi_a}^{\phi_b} T\,d\phi = \text{area } abnm$$

* It is particularly important not to confuse this word *uniform* with the word *constant*.

For the condition given above, if the heat had been added at constant temperature, as from b to f, Fig. 1b, then,

$$H\Big]_b^f = \int_{\phi_b}^{\phi_f} T\, d\phi = T\Big[\phi_f - \phi_b\Big] = \text{area } bfkn$$

or, $\phi_f - \phi_b = \dfrac{H]_b^f}{T}$ = distance bf or nk.

For both of these cases the addition of heat to the substance caused an increase of entropy. Had the heat been abstracted, the entropy would have been decreased. Note that it is always *the change* in entropy that engineers are interested in, rather than its absolute values.

Adiabatics.—The term *adiabatic* means no transfer of heat. Hence an adiabatic expansion or compression means one in which no heat is added or abstracted during the process.

Reversible and Irreversible Adiabatics.—The two general classes into which all adiabatics may be divided, are called reversible and irreversible adiabatics. An adiabatic expansion or compression, which is frictionless and which takes place in such a manner that the substance passes through a continuous series of uniform states, would be called a *reversible adiabatic*. A reversible adiabatic is also called a constant entropy line or isentropic, because a vertical line on the temperature-entropy chart is the only one which permits the area underneath it to become equal to zero, thus satisfying the definitions of adiabatics and entropy.

It is impossible to have these conditions fulfilled in actual cases, but it is useful to study them and compare them with the actual conditions which may be made to approach very close to the ideal. Thus, a substance may expand in a single cylinder, with but very little friction, and with but very little transfer of heat to or from the cylinder walls, and slowly enough so that all of the substance in this cylinder is almost exactly in the same state. Such cases may properly be treated as reversible adiabatics or isentropics.

On the other hand, sudden or "free" expansions are sure to set up eddies whose kinetic energy will soon reappear in the form of heat, thus causing an increase of entropy. Friction will have a

similar effect. Such expansions are called irreversible, and if they have taken place without any heat being transferred to or from another substance, they would be called *irreversible adiabatics.*

In the case of adiabatic expansion of gases or steam through a properly-formed nozzle in which there are no eddies formed, and in which there is no friction, the transformation of the available heat energy into velocity is complete and may therefore be considered as a reversible adiabatic. In actual steam turbine nozzles the loss due to friction and eddies is extremely small, so that in turbine design the expansion in the nozzle is justly assumed to be a reversible adiabatic. The main losses in the turbine occur in the endeavor to transfer the energy of the jet to the turbine blades. These losses cause considerable reheating, so that in any actual case there is considerable increase in the entropy of the steam in passing through the turbine. As an illustration, see problems 41 and 42.

The term adiabatic will be used hereafter to mean reversible adiabatic, unless specifically stated otherwise.

Work and Heat Depend upon Path.—From the consideration of the PV and Tϕ charts it will be evident that it is not sufficient to give merely the initial and final states of a substance in order to find the work done and the heat required to go from one state to another. It is also necessary to give the exact path to be followed. Thus, in Figs. 1a and 1b, heat might be added and abstracted in such a manner that the state *c* is finally reached by means of the constant pressure line *ab*, and the constant volume line *bc*, or this same state might have been reached more directly by some such line as *adc*.

By observing the areas it will be seen that the work done in the first case is *abhg*, while for the path *adc* the work done is represented by the area *adchg*. Also from the Tϕ chart it may be seen that in going from *a* to *b* an amount of heat equal to the area *abnm* was supplied to the substance, and in going along the constant volume line, *bc*, an amount of heat equal to the area *bcmn* was abstracted from the substance. The substance might also have reached the state *c* by expanding adiabatically along the line *adc;* but note the difference in areas representing the work and heat.

Cycles.—If a substance should be made to follow a series of paths such as *ab*, *bc*, and *ca*, so that it is returned to its initial state, the substance would then have completed a cycle, and the substance used would have been called the working substance. Had the cycle been completed in the order of the letters, *abc*, an amount of work in foot-pounds, equal to the area, *abc*, Fig. 1a, would have been done upon some external mechanism. The heat equivalent in B. t. u. of this work, would be represented by the area *abc*, Fig. 1b. This heat would have been supplied to the working substance from some external source.

Properties which are Independent of the Path.—After a cycle has been completed, the substance is in exactly the same condition as at the beginning, so its pressure, volume, temperature, entropy, and intrinsic heat must all be the same as originally. Likewise, if a substance is in a certain state, such as *c*, it merely means that *all of the above properties have some definite value and may be determined regardless of the manner in which this substance may have reached this state c.*

Intrinsic Heat or Intrinsic Energy.—Either of these terms may be used to represent all of the heat energy contained within a substance measured above some convenient standard. It is to be carefully noted that this is a different quantity in general from the heat required to bring the substance to a particular state from the standard condition. This is on account of the fact that, when heat is added to a substance, in which no heat is lost by radiation or otherwise, in which there is no change in electrical or chemical energy, or in which there is no change in the kinetic energy of the substance due to its mass velocity, then in going from *a* to *b*, a general equation may be written, thus:

$$\left.\begin{matrix}\text{Heat}\\ \text{added}\end{matrix}\right]_a^b = \left.\begin{matrix}\text{Gain in intrinsic}\\ \text{heat}\end{matrix}\right]_a^b + \left.\begin{matrix}\text{External work}\\ \text{done}\end{matrix}\right]_a^b$$

The external work done means that work is done by the substance upon some external mechanism. In case this term is negative, it means that the work is done by the external mechanism upon the substance.

Steam Formed at Constant Pressure.—For nearly all commercial purposes steam is generated in a steam boiler at a pressure which is maintained nearly constant. It is therefore very important to consider carefully the formation of steam at constant pressure. It has been found by experiment that during the change of state of any substance from a liquid to a vapor, that if the pressure be kept constant the temperature of the vapor in contact with its liquid will also remain constant until the vaporization has been completed.

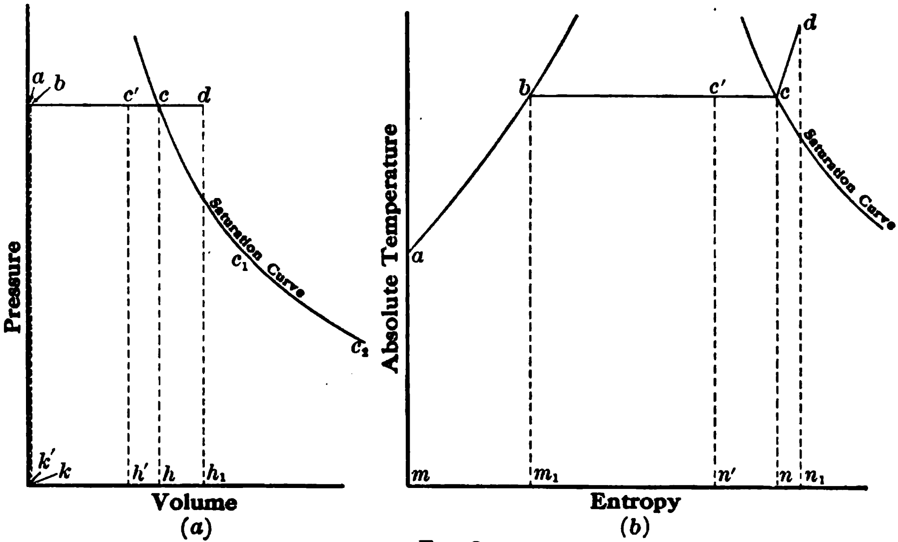


FIG. 2.

Suppose that a pound of water at a temperature of melting ice, 32° F., is contained in a metal cylinder with a tight-fitting piston resting upon the surface of the water. The area of this piston and the total weight resting upon the water being known, the absolute pressure on the water may be determined. Let this value in pounds per square inch be denoted by p, and in pounds per square foot by P. The volume of this pound of water will be very small, only .016 cubic feet. We may represent this starting point by the letter *a* in Figs. 2a and 2b. (We locate the point *a* on the zero entropy line merely as a matter of convenience, as we shall be concerned only with the heat measured above 32° F.)

Let heat be applied to the cylinder now, and it will be found that at first the piston moves up only a little, due to the slight increase in the volume of the water. The temperature, however, is observed to rise rapidly until it reaches some definite point dependent entirely upon the pressure imposed by the piston. Let this state be represented by *b*. The substance is now all in the form of liquid, but at the particular temperature known as the temperature of vaporization for this given pressure.

Since the volume has changed only very slightly, the points *a* and *b* on the PV diagram will almost exactly coincide. Even though this water had been heated to a temperature of 400° F., as it might have been if under a pressure of 250 pounds or more, its volume would still only be .0187 cubic foot. This extremely small change in volume means that very little work, shown by the area *abkk'*, Fig. 2a, has been done on the piston, and consequently all the heat that has been applied has been used to increase the sensible heat of the water.

The Heat of the Liquid is the term applied to this heat, which is used to heat a unit weight of water from 32° F. to the temperature of vaporization. The area representing the heat of the liquid is abm_1m, Fig. 2b. With further addition of heat it will now be found that the volume is increased very much and that the temperature remains constant until some state *c* is reached. The water has now all been evaporated, and exists in the form of dry saturated steam. Any further addition of heat will cause it to become superheated, and any abstraction of heat will cause it to be partially condensed. The heat which was necessary to vaporize this pound of water under this pressure is represented by the area $bc\,n\,m_1$, and is called *the latent heat of vaporization*, or latent heat of steam.

The Total Heat of Dry Saturated Steam is equal to the heat of the liquid plus the latent heat of vaporization and is equal to the area *abcnm*, Fig. 2b.

The Saturation Curve on any diagram is the curve which shows the two coordinate values of dry saturated steam for the entire range of scales on that diagram. Thus in Fig. 2a the specific volume of dry saturated steam may be determined for any pressure by

means of the saturation curve, $c\,c_1\,c_2$. The saturation curve may also be seen for other coordinates as in Fig. 2b, Plate 1b, or Plate 8a.

Wet Steam and Quality.—During the formation of a pound of dry saturated steam in the manner just outlined, it might have been possible to stop the addition of heat before all the water was evaporated and the resultant mixture of water and steam would be what is known as wet steam. That portion by weight of this mixture which is dry vapor would be known as the *Quality* of the steam. Thus if x represents quality and if nine-tenths of the pound of water have been vaporized, this fact would be expressed by the equation $x = 90\%$. In Fig. 2b the quality at *c* is unity and at *b* the quality is zero; or $x_c = 100\%$ and $x_b = 0$. It is important to note that this interpretation of quality is the most useful one in using the $T\phi$ chart. In this chart it is understood that a unit weight of substance is being considered unless it be definitely stated otherwise, and it is sometimes necessary to carry this unit weight of substance throughout the extreme range of quality in order to analyze a cycle or to understand many important thermodynamic relations.

On the other hand, steam coming from a boiler often carries with it a considerable amount of moisture. If a sample of this mixture be obtained, then the weight of the dry steam present divided by the weight of this entire mixture would be the quality of the steam coming from the boiler, the weight of water still remaining in the boiler not being considered at all in this determination of quality.

The Total Heat of Wet Steam is equal to the heat of the liquid plus the quality multiplied by the latent heat of vaporization, or $h + xL$, where h represents the heat of the liquid and L the latent heat. If c′, Fig. 2b, represents this state of the steam, then the area *abc′n′m* is equal to the total heat at c′.

Superheated Steam.—If heat is added to dry saturated steam, the pressure remaining constant as before, the volume and temperature will be found to increase; and the steam is said to be superheated. Superheated steam may therefore be defined as any steam, regardless of how formed, having a temperature higher than

the temperature of saturated steam of the same pressure. The difference between these two temperatures is called the *Degrees of superheat,* and is nearly always represented by the letter D. For the state *d,* Fig. 2b, $D = T_d - T_c$. If the *specific heat of superheated steam* at constant pressure be represented by c_p, then c_p D is equal to the area cdn_1n and is called by either of the terms, *heat of superheat, heat of superheating* or *superheat.*

The value of the specific heat of superheat steam is variable, and depends upon both the pressure and the degrees of superheat.

The Total Heat of Superheated Steam is equal to the total heat of dry saturated steam plus the heat of superheat. Thus for the state point *d,* Fig. 2b, the total heat at *d* is equal to the area $abcdn_1m$.

Total Heat may now most conveniently be defined in the manner in which it is used by engineers, that is, a general definition which will hold for wet, dry, or superheated steam.

Total Heat of Steam in any Given State is the amount of heat required to heat at constant pressure a unit weight of water from the temperature of melting ice to the state under consideration.

Thus referring to Fig. 2b:

$$\text{Total Heat}\Big]_{c'} = \text{area } abc'n'm = h + x_c' L$$

$$\text{Total Heat}\Big]_{c} = \text{area } abc\,n\,m = h + L$$

$$\text{Total Heat}\Big]_{d} = \text{area } abcdn_1m = h + L + c_pD.$$

In the United States the units are almost always B. t. u. per pound. All values of total heat which are likely to be needed will be found on Plates 1 to 7 inclusive.

External Work at Constant Pressure.—During the period of vaporization the volume of a pound of water is changed to the very much larger volume of dry saturated steam, as from V_b to V_c, Fig. 2a.

If this volume is represented in cubic feet by V and the pressure

in pounds per square foot by P, the work done during vaporization is

$$\text{Work}\Big]_b^c = P\Big[V_c - V_b\Big] \text{ ft.-lbs.} = \text{area } b\,c\,h\,k,$$

$$\text{or} = AP\,(V_c - V_b) \text{ B. t. u.}$$

A being taken as the reciprocal of the mechanical equivalent of heat, or $A = \frac{1}{778}$ or $\frac{1}{777.5}$

$AP\,(V_c - V_b)$ is commonly known as the *external latent heat of vaporization.*

During superheating the external work done is

$$\text{Work}\Big]_c^d = P\Big[V_d - V_c\Big] \text{ ft.-lbs.} = \text{area } cdh_1h$$

$$\text{or} = AP\,(V_d - V_c) \text{ B. t. u.}$$

For the entire process starting with water at 32° when its specific volume would accordingly be .016 cubic feet per pound, the external work done in order to reach some state such as c′ or d would be

$$\text{Work}\Big]_a^c = P\Big[V_c' - V_a\Big] \text{ ft.-lbs.} = AP\Big[V_c' - .016\Big] \text{ B. t. u.}$$

$$\text{Work}\Big]_a^d = P\Big[V_d - V_a\Big] \text{ ft.-lbs.} = AP\Big[V_d - .016\Big] \text{ B. t. u.}$$

This is what is called *the constant pressure external work,* and is given for all conditions of steam by Plates 8a and 8b.

This work has been done upon the piston, so that after some state such as *d* has been reached, the piston and all weights resting upon it have more potential energy by the amount of $AP\,[V_d - V_a]$ B. t. u. than they had at the beginning of application of heat to the water in condition *a*. This increase in potential energy of the piston and its weights has come from the heat which was necessary to form the pound of steam along the constant pressure path *abcd*. Now, if from the total amount of heat which has been added to a substance in order to reach a certain state by going along a certain path, all of the heat which has been used to do external work be subtracted, the amount left would be the gain in intrinsic energy of the substance, provided there were no losses

and that no energy had been used to impart velocity to the substance.

The Intrinsic Energy, Internal Energy, or Intrinsic Heat of steam, then, merely means the heat energy contained within the steam above 32°, and may always be found by subtracting the constant pressure external work from the total heat. The total heat may be obtained directly from Plates 1 to 7, while the external work may be obtained from Plates 8a and 8b. Since we can not reduce the temperature of any substance to absolute zero, it is not possible to reach the state in which the intrinsic heat is zero. For work with steam we need only be concerned with the intrinsic heat above 32° F.

Specific Volume of steam means the volume of one pound of the steam in its given state. Unless some quality or superheat is given, it is understood to refer to dry saturated steam.

The specific volume of wet steam may be determined by means of the quality in the following manner:

Let $V_{sat.}$ = volume of 1 lb. of dry saturated steam of given pressure,

V_w = volume of 1 lb. of water at the temperature of vaporization,

= .016 cu. ft. at 32° F.,

= .017 cu. ft. at 250° F.,

= .018 cu. ft. at 350° F.,

= .019 cu. ft. at 420° F.

Let u = change of volume during vaporization,

$= V_{sat.} - V_w$.

Then for some point such as c′ having a quality x_c', the specific volume is

$$\begin{aligned} V_c' &= V_w + x_c'u, \\ &= V_w + x_c'\,(V_{sat.} - V_w), \\ &= (1 - x_c')\,V_w + x_c'\,V_{sat.} \end{aligned}$$

This becomes equal to $x_c'\,V_{sat.}$ almost exactly, except when the quality is very low or the pressure is higher than atmospheric.

The specific volume may be read directly from Plates 1 to 8 inclusive.

Entropies.—For convenience, entropy of water at 32° is taken as zero. Hence the *entropy of the liquid* per pound of water, from Fig. 2b, is

$$\phi_b = \int_a^b \frac{dH}{T} = \int_{T_a}^{T_b} \frac{c_p\, dT}{T}$$

Where c_p = the specific heat of water at constant pressure. This value depends upon the temperature, so the integration is not often made, but instead the entropy of the liquid is usually obtained from the steam tables.

The Entropy of Vaporization is, from Fig. 2b,

$$\phi_c - \phi_b = \int_b^c \frac{dH}{T} = \frac{L}{T_b} = \frac{\text{Latent heat}}{\text{abs. Temp.}}$$

The Entropy of Superheat or the change of entropy due to superheating is, from Fig. 2b,

$$\phi_d - \phi_c = \int_c^d \frac{dH}{T} = \int_{T_c}^{T_d} \frac{c_p\, dT}{T}$$

where c_p represents the constant pressure specific heat of superheated steam.

The Entropy of Steam in any state means the total entropy up to that state, measured above the assumed zero of entropy.

For Wet Steam the Total Entropy is equal to
Entropy of the liquid + (Quality) (Entropy of vaporization)

For Superheated Steam the Total Entropy is equal to
Entropy of the liquid + Entropy of vaporization + Entropy of Superheat

The Total Entropies are plotted on Plates 1 to 7 inclusive and these should serve nearly every purpose for which the engineer has need of their numerical values.

Rankine and Clausius Cycles.—The most useful cycle in power-plant work is the one represented by *abfe*, Figs. 5 and 6, Problem 29. This cycle is sometimes known as the Clausius Cycle and sometimes as the Rankine Cycle, but since the analysis of this cycle was

first published by each of these men at the same time, there will probably always be a difference of opinion as to which name should be used.

If, instead of allowing the adiabatic expansion to continue until the back pressure is reached, as at *f*, the expansion had been stopped at the point *c*, thus cutting off the "toe" in each diagram, there would be another cycle, *abcde*, which is preferred by many engineers as a basis of comparison for the performance of reciprocating engines. This latter cycle is oftentimes called the Rankine and the other the Clausius, thus making a convenient distinction. They are also spoken of as the "complete expansion" and the "incomplete expansion" cycles. These latter terms will be used in the problems of this book in order that there may be no possible confusion of the two cycles by using the terms Rankine and Clausius.

Available Energy is a term that may have many meanings, but when used in connection with steam cycles it means the energy of the steam which would be converted into work by an ideal mechanism that would carry out exactly the theoretical cycle. Thus, in using Table IV, the engineer would consider the term to represent the net area of the complete expansion cycle, by assuming that all of this energy is available for the production of velocity in the ideal nozzle. Then with an ideal turbine all of this velocity energy could be transformed into work. When designing a multi-stage turbine, however, it is necessary to find the energy available for each stage. This amount of energy is affected by the reheating in each of the preceding stages, as well as the drop in pressure in the stage being considered.

For any steam prime mover, the available energy is a small part of the heat supplied to it, since a very large part of this heat must necessarily be given up to the exhaust.

PREPARATION AND USE OF THE STEAM CHARTS AND TABLE OF VELOCITIES

The general idea of the main diagram, which consists of Plates 1 to 6, inclusive, may be readily obtained from the index chart. It is seen to be divided into twelve equal parts, the top and bottom halves of each section being indicated by the subscripts *a* and *b*, respectively. These two halves will be found facing each other, so that wherever this chart is opened, a complete section may be seen.

The same total heat scale is used throughout, but the volume scale for each section is changed so that the general relation of each family of curves to one another will remain about the same.

Having established the scale of volumes and total heats, constant pressure lines were then plotted from the values given by the steam tables of Marks and Davis. In the superheated region these lines are slightly curved, but in the wet region they are straight. For those pressures not given in the steam tables—all fractional pressures and those given in inches of mercury—the volumes and total heats were determined in two ways. For the wet region, special auxiliary curves were drawn by computing for certain qualities the total heats and volumes for those pressures given by the steam tables. From such auxiliary curves the desired values could then be determined, and the curves plotted in their correct relative positions. In the same manner the corresponding values were found for all fractional pressures above one pound in the superheated region. For those pressures less than one pound the volumes in the superheated field were found from Linde's equation and the total heats were determined by adding c_pD to the total heat of the dry saturated steam. The values of the specific heat for these low pressures were determined by assuming as correct the specific heats used by Marks and Davis for the pressures from one to four pounds. A curve of specific heats was then constructed through these points and extended into the region of lower pressures. From such a curve the specific heats were read. This was the best method by which the lines in the superheated region of

Plate 6a could be drawn so that they would all continue as smooth curves.

For the region of very low pressures, such as those given on Plate 6a, it was found that constant temperature lines would almost coincide with the lines of constant total heat when in the superheated region. It was, therefore, considered worth while to put on the scale of approximate temperatures as given on the upper right-hand corner of this sheet. They will be found to be of service when it is desired to determine the pressure corresponding to a certain temperature and specific volume. For a large part of this superheated region this temperature scale agrees with the temperature as obtained from the degrees of superheat and the temperature of vaporization. It is not intended, however, to be used as an accurate means of obtaining the temperature, as the error in using it may easily be one degree.

Entropies.—To draw accurately the lines of constant entropy it was necessary to construct a large total heat-entropy diagram from which could be obtained the values of total heat for the various pressure lines and any entropy line. In the superheated region this was done for each entropy line, but for the wet region only every fifth line was obtained in this manner, as the others could be put in by dividers.

Qualities.—The lines of constant quality were obtained by computing the total heats for such qualities as 70, 80, and 90 per cent., and after these were plotted, the intermediate ones were obtained by dividers.

The Heat of the Liquid Curve.—It was desired to have the heat of the liquid for all pressures, if it could be obtained without interfering with the other lines of the chart. The lower left-hand portion of each section was the only space available, and it was found by trial that this curve would fit there very conveniently. Since it is important to be able to read the heat of the liquid just as accurately as the total heat, these values should naturally be plotted on scales of equal magnitude. This has been done by supplying the numbers in parentheses, thus establishing the heat of the liquid scale.

Temperature of Vaporization.—Since the temperature of dry saturated steam is the same as that of wet steam having the same pressure, it is possible to construct a scale on any line intersecting the constant pressure lines in the wet region, so that such a scale will represent the temperatures of vaporization. Inasmuch as the heat of the liquid is often desired for some definite temperature, it is natural to try to place these two curves as close together as possible. They were therefore combined by merely graduating the heat of the liquid curve, already drawn, so that it would also give the temperature of vaporization for each pressure.

In order to find the temperature of the superheated steam, it is merely necessary to add the degrees of superheat to the temperature of vaporization.

Plate No. 7 is a special addition to the main heat-volume chart in order to enable one to work with unusually high superheats for pressures ranging from 10 to 45 pounds per square inch absolute, such as are used when reheating steam to superheats of 500 or 600°, as is done in the Ferranti turbine. It was not thought worth while to show this plate on the Index Chart.

Plate No. 8 gives the external work in B. t. u. done by a pound of steam during its formation at constant pressure from water at 32° F., until it reaches the state under consideration. For Plate 8a there are two volume scales and for 8b four are used. The small drawing in the upper right-hand corner is intended to give the external work for the low pressures when the quality is relatively high, so that the volumes become too great for the main part of 8b. For pressures below one pound absolute, there is a region for which no values are given, but it will seldom, if ever, happen that such values will be desired.

Table of Velocities.—Any body having a weight of one pound and a velocity of v feet per second, will have kinetic energy due to this velocity equal to

$$\frac{v^2}{2g} \text{ or } \frac{v^2}{64.34} \text{ ft.-lbs.}$$

which is equal to

$$\frac{v^2}{64.34 \times 777.5} \text{ B. t. u.}$$

Calling this kinetic energy in B. t. u., E, the equation becomes

$$v = \sqrt{64.34 \times 777.5\,E}$$
$$= 223.7\ \sqrt{E}$$

Then, for a nozzle which transforms the available heat energy into velocity without any losses, the velocity is

$$v = 223.7\ \sqrt{\text{available energy, B. t. u.}}$$

Table IV was prepared by using this equation.

The available energy in a steam nozzle is usually taken as the total heat at entrance to the nozzle minus the total heat for the nozzle back pressure, and the same entropy as at entrance. This neglects the small difference between $P_1 V_{w1}$ and $P_2 V_{w2}$ where P represents pressure and V_w the volume of the water. This difference is usually far too small to be considered in actual cases. Thus, taking .017 as the average volume in cubic feet of a pound of hot water, this difference is equal to

$$\frac{.017 \times 144}{778} (p_1 - p_2) = .00315\ (p_1 - p_2) \text{ B. t. u. per lb.}$$

For a drop of 100 pounds per square inch in passing through the nozzle, this difference would, therefore, become only 0.3 B. t. u. per pound.

Using the Charts.—Whenever any two of the properties of superheated steam are given, that is sufficient to locate the state point on the chart from which all of the other values may be determined at once. This is also true for wet steam, except for the case in which the two values given are pressure and temperature. These two alone are not sufficient to determine the state point, since the temperature of wet steam is independent of its quality. The problems will supply many illustrations of the use of the charts and tables.

ATMOSPHERIC PRESSURE AND BAROMETRIC CORRECTIONS

The average pressure of the atmosphere at sea level is about 14.7 pounds per square inch, and for many purposes this value is all that is needed. On the other hand, it is often necessary to determine carefully the atmospheric pressure at the time and place desired. This is usually done by measuring the height of a column of mercury which is just balanced by the atmosphere. Pressures less than atmospheric are also often determined by means of mercury columns. All such measurements may be accurately made, if care is exercised in obtaining the readings and in applying the proper corrections.

The Standard Atmospheric Pressure * is equivalent to the height of a column of mercury 760 millimetres high, at a temperture of 0° C., at sea level and 45° latitude. Reducing this to the inch basis the standard atmospheric pressure becomes

$$760 \times .03937 = 29.9212 \text{ inches}$$

at 32°, at sea level and latitude of 45°. Since the engineer is not concerned with any readings of pressure closer than the thousandth of an inch of mercury, and very often only to the nearest hundredth, this is equivalent to the value usually given, viz., 29.921 inches at 32° F.

The Thirty-Inch Barometer.—When measuring the pressure in a condenser, engineers often speak of the vacuum referred to a thirty-inch barometer. The meaning of this expression may not always be the same, as it may be interpreted differently. However, a logical meaning and one quite generally used † is that 30 inches

* The exact value adopted for international use by the Third General Conference on Weights and Measures is that of a column of mercury 760 millimetres high, the mercury being at a temperature of 0° C. and the acceleration of gravity being 980.665 centimetres per second per second. See Vol. XII, p. 66, of the "Travaux et Mémoirs du Bureau International des Poids et Mesures."

† See Genhardt's "Steam Power Plant Engineering," p. 468, 4th edition. Also "Steam Tables for Condenser Work" by the Wheeler Condenser and Engineering Company, page 5.

of mercury at some definite temperature will mean the same pressure as do the 29.921 inches at 32° F. To find this temperature the equation

$$30.00 = 29.921 \left[1 + a\ (t - 32)\right]$$

may be used where
t = the required temperature, and
a = .000101 = coefficient of cubical expansion of mercury per degree F.

From this equation

$$t = 58.1° \text{ F.}$$

Had the standard pressure been taken as 29.92 inches instead of 29.921 inches, this temperature would, from the above equation, become 58.4° F.

In making the above correction for temperature, it was assumed that the scale by which the mercury is measured is correct at this temperature. This is often the case, as when very short scales or scales made of non-expansive materials are used. However, full-length brass scales, or their equivalent, are sometimes used. If such a brass scale be correct at 62° F., as is usually intended, the temperature at which 30 such inches represent the standard pressure may be found from the equation:

$$30.00 = 29.921 \left[1 + a\ (t - 32) - b\ (t - 62)\right]$$

where a = .000101 as above,
and b = .00001 = coefficient of expansion of brass.

The solution of this equation will give, t = 57.6° F.

The difference in the above three values for this temperature is not of great importance. In all the steam charts used in this book the temperature of the mercury has been taken as 58.1° F., as that would seem to serve a more general use than does the value 57.6° derived on the assumption of a full-length brass scale having a definite coefficient of expansion.

Correction Due to Temperature.—In order to facilitate the reduction of readings of a mercury column at any temperature to the temperature of 58.1°, Plate 9a has been prepared. It is correct

for those cases in which a non-expansive scale is used, and may also, of course, be used indirectly to correct to any other temperature desired, say 32°.

When a Full-Length Brass Scale is used to measure the height of a column of mercury of some considerable magnitude, such as a barometer, and the temperature of this scale is not close to 62°, the correction should be made to the scale as well as to the mercury.

Thus, for barometers having such scales, it may be desired to reduce the reading to 32° for the mercury and 62° for the brass scale. The following values * give this correction for a column 30 inches high. Hence, for any other column, say h, it is only necessary to multiply by $\left(\frac{h}{30}\right)$

Temp.° F.	Correction in Inches (For 30 inches)	Temp.° F.	Correction in Inches (For 30 inches)
100	Subtract .193	40	Subtract .031
95	.180	35	.017
90	.166	32	**.009**
85	.153	30	.004
80	.139	**28.5**	*Add* **.000**
75	.126	25	.010
70	.112	20	.024
65	.099	15	.037
62	**.091**	10	.051
60	.085	5	.065
58.1	**.080**	0	.078
55	.072	−5	.092
50	.058	−10	.106
45	.045	−15	.119

Variation of Atmospheric Pressure with Altitude.—It may happen that it is not convenient or practicable to have a barometer at the same place at which it is desired to measure the pressure of the atmosphere. It does not require much of a difference in elevation in

* From Table 1, "Circular F, U. S. Weather Bureau No. 472."

order to make quite a variation in atmospheric pressure. To obtain the true pressure at some elevation which is not extremely different from that at which a barometer is read, it is only necessary to observe the temperature of the atmosphere and obtain the altitude of each place. Then, by means of Plate 9b, this correction may be readily obtained. This plate* was intended to serve only in those cases in which the change in altitude might be 100 feet or less. But after completion it was found to agree with the corrections obtained from the above table for variations in altitude of as much as 1,000 feet. The average altitude is to be used when obtaining the correction from this plate.

Reduction to Standard Gravity.—When it is desired to make a true comparison of pressures determined by heights of mercury columns, it is necessary to reduce them to a common standard of gravity. The standard usually adopted for this is that at sea level and a latitude of 45°. Table I† will give these corrections for the various latitudes.

The variation of gravity due to change in altitude is usually much too small to require correction of mercury columns. This correction is proportional to the height of the mercury and for a 30-inch column amounts to about .0018-inch for each 1,000 feet above sea-level. It is, of course, to be subtracted. It is important not to confuse this correction with the variation of the pressure of the atmosphere itself, due to change in altitude, as has already been given.

Correction Due to Capillarity.—The capillary action between mercury and glass will cause a depression of the column if the pressure is constant so that the mercury becomes stationary. The correction to be made for this action may be very large in case the tube or the mercury is dirty or if a very small tube is used. The height of the meniscus and the diameter of the tube are the two most important factors required to determine the amount of this correction. Table II will give these corrections for nearly all cases.

* Prepared from Table 21, Vol. II, "Report of the Chief of the Weather Bureau, 1900–1901."
† Abridged from Table 101 of the "Smithsonian Tables."

Reduction of Mercury Column to Pounds per Square Inch.—For the most important work in connection with the measurement of steam pressures by means of mercury columns, it is not usually necessary to convert the reading to any other unit. Thus the absolute pressure in a condenser being measured by the difference in the heights of two mercury columns, it is convenient and proper to use the inch of mercury as the unit to express such pressures.

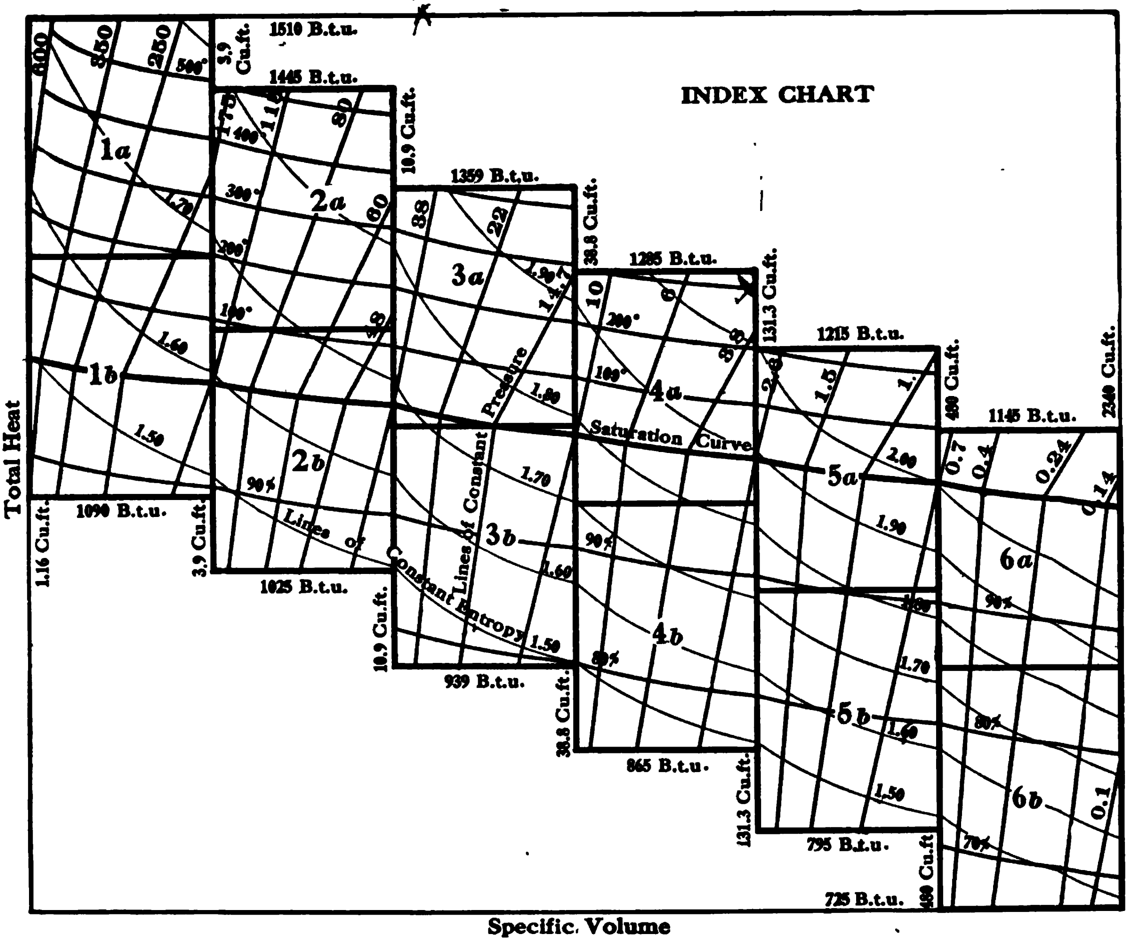
Density of Mercury.—When it is necessary to convert inches of mercury to pounds per square inch, Table III may be used. This table was prepared by using the following equation:

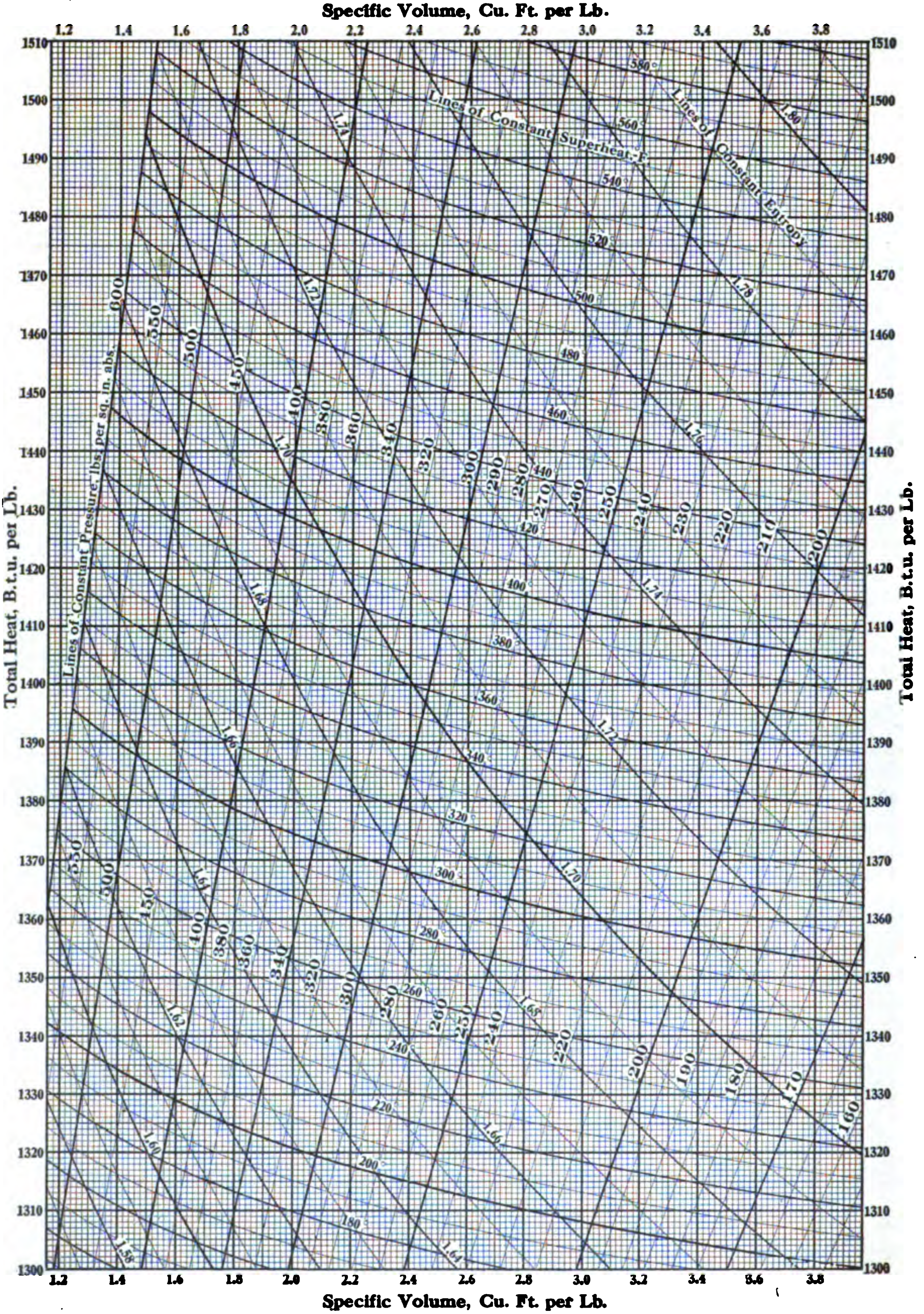
One cubic inch at 32° F. is equal to

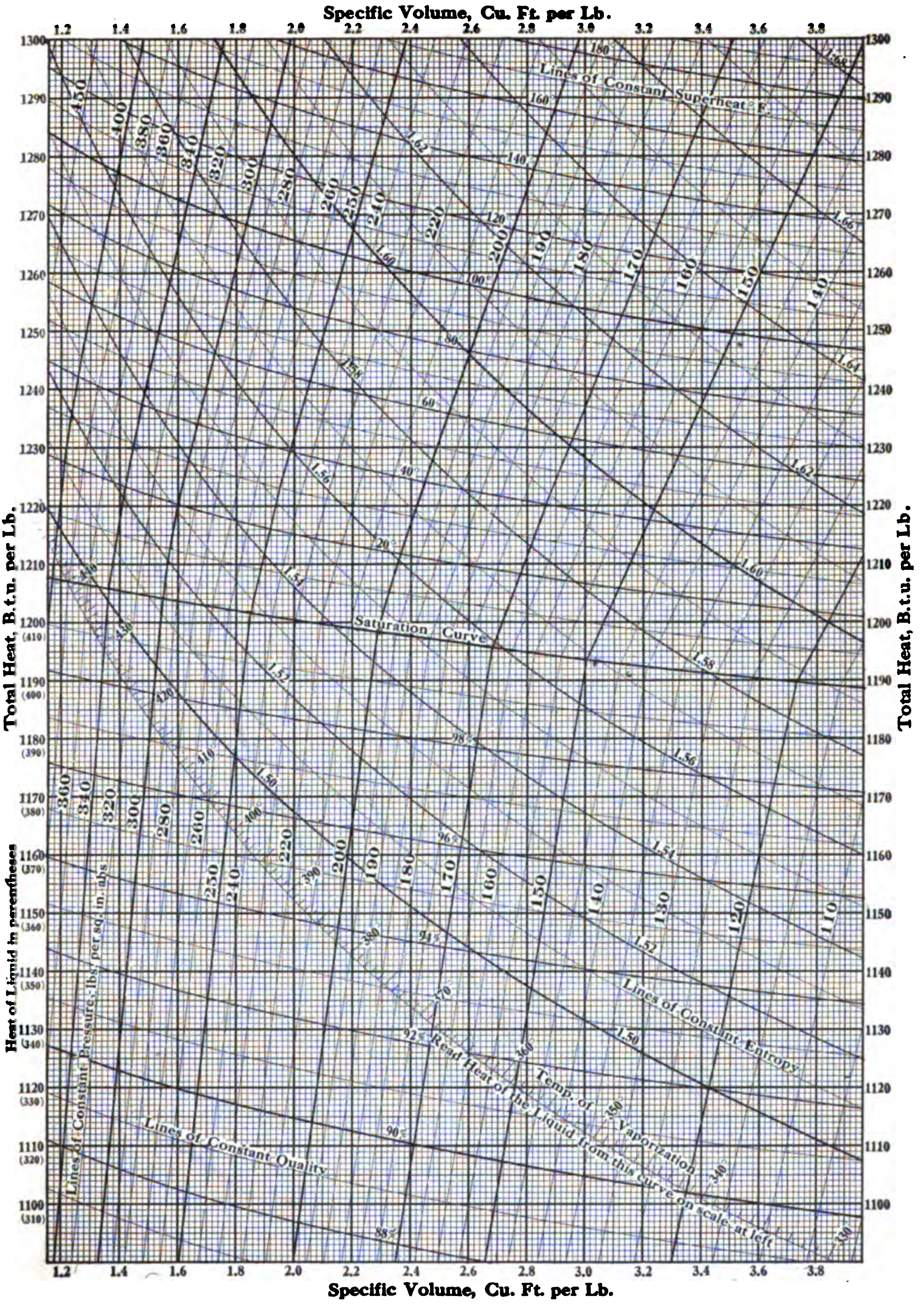
$$1 + (.000101)(t - 32) \text{ cu. in. at } t° \text{ F.}$$

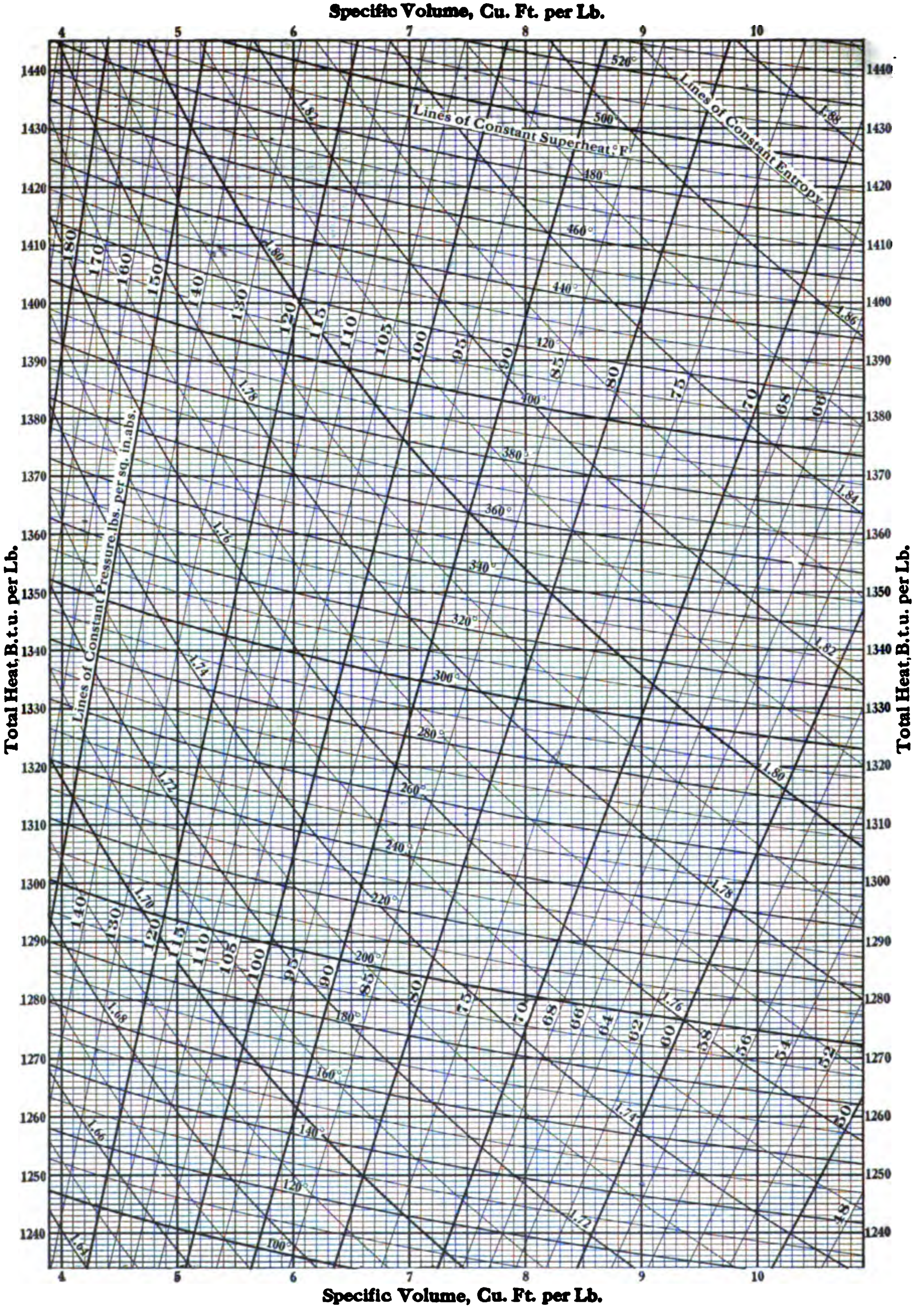
The density, 0.49117 pounds per cubic inch at 32° F., was obtained from the equivalent, 13.59545 grams per c.c. at 0° C.* Then these values for all temperatures were afterward compared and found to agree with Table 76 of the Smithsonian Tables.

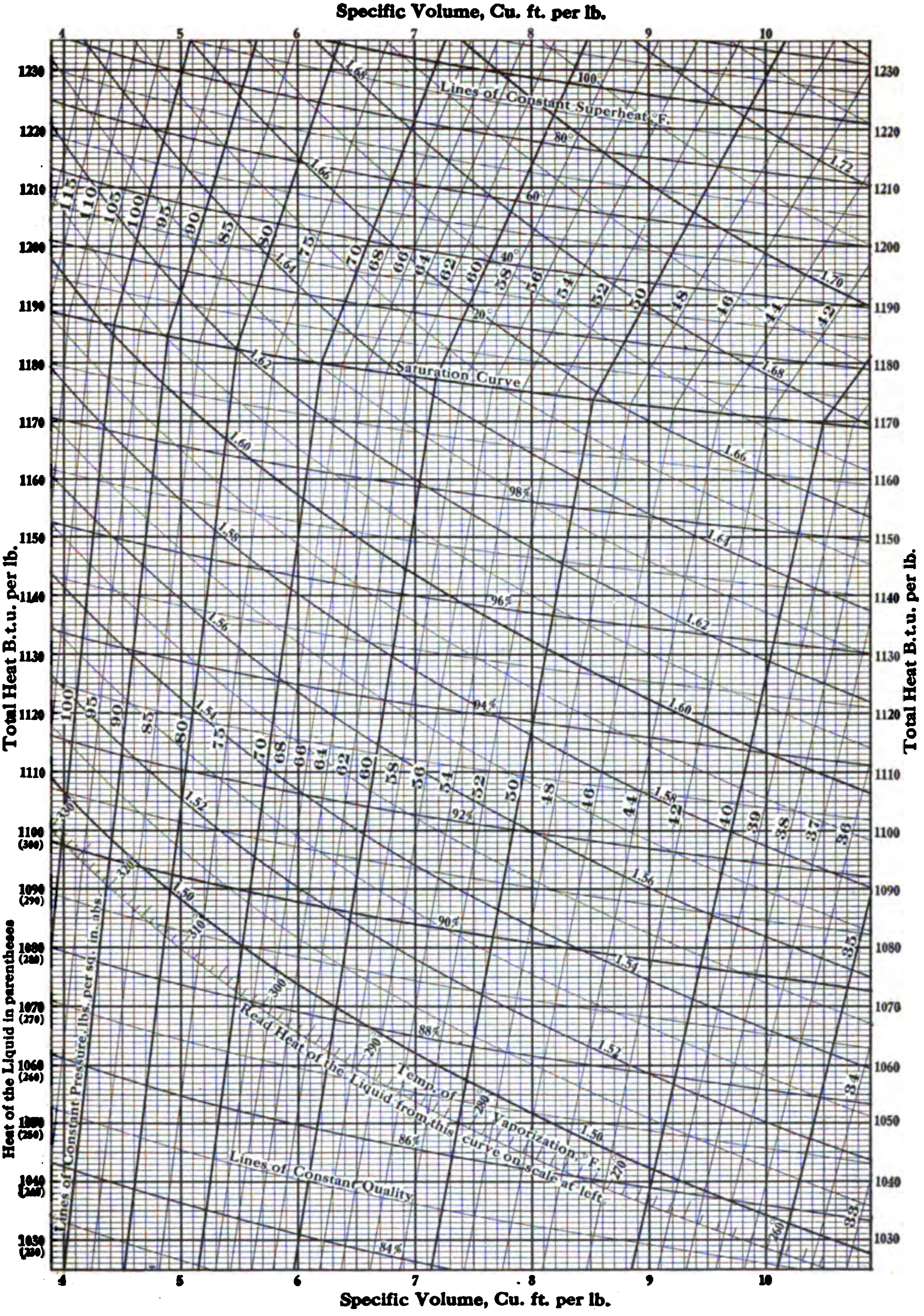
* This is the value given by Table 19 of "Landolt and Börnstein Physikalisch-Chemische Tabellen."

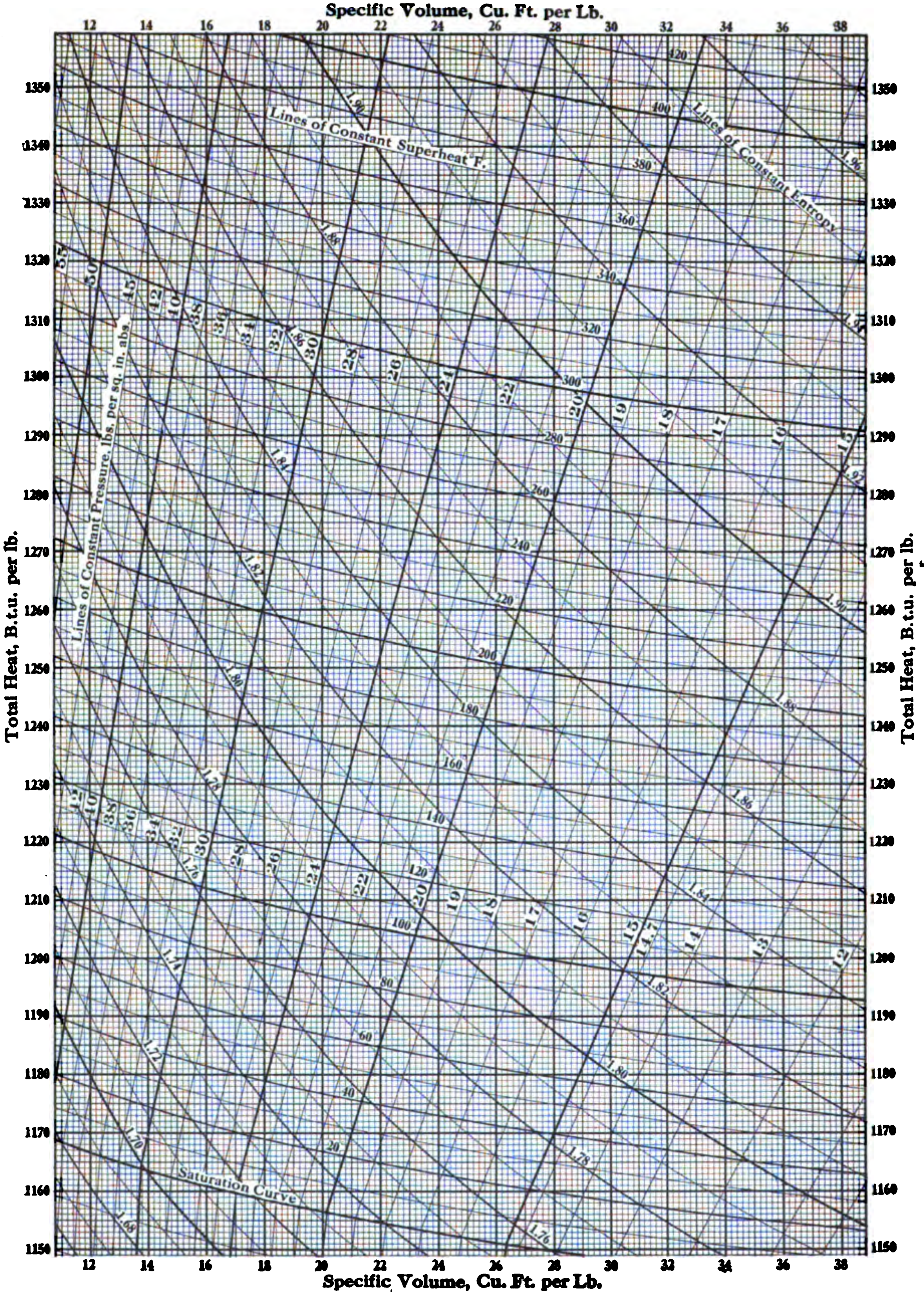

INDEX CHART
Total Heat
Specific Volume
Saturation Curve
Lines of Constant Pressure
Lines of Constant Entropy
1a
1b
2a
2b
3a
3b
4a
4b
5a
5b
6a
6b
1510 B.t.u.
1445 B.t.u.
1359 B.t.u.
1285 B.t.u.
1215 B.t.u.
1145 B.t.u.
1090 B.t.u.
1025 B.t.u.
939 B.t.u.
865 B.t.u.
795 B.t.u.
725 B.t.u.
1.16 Cu.ft.
3.9 Cu.ft.
10.9 Cu.ft.
38.8 Cu.ft.
131.3 Cu.ft.
480 Cu.ft.
2340 Cu.ft.

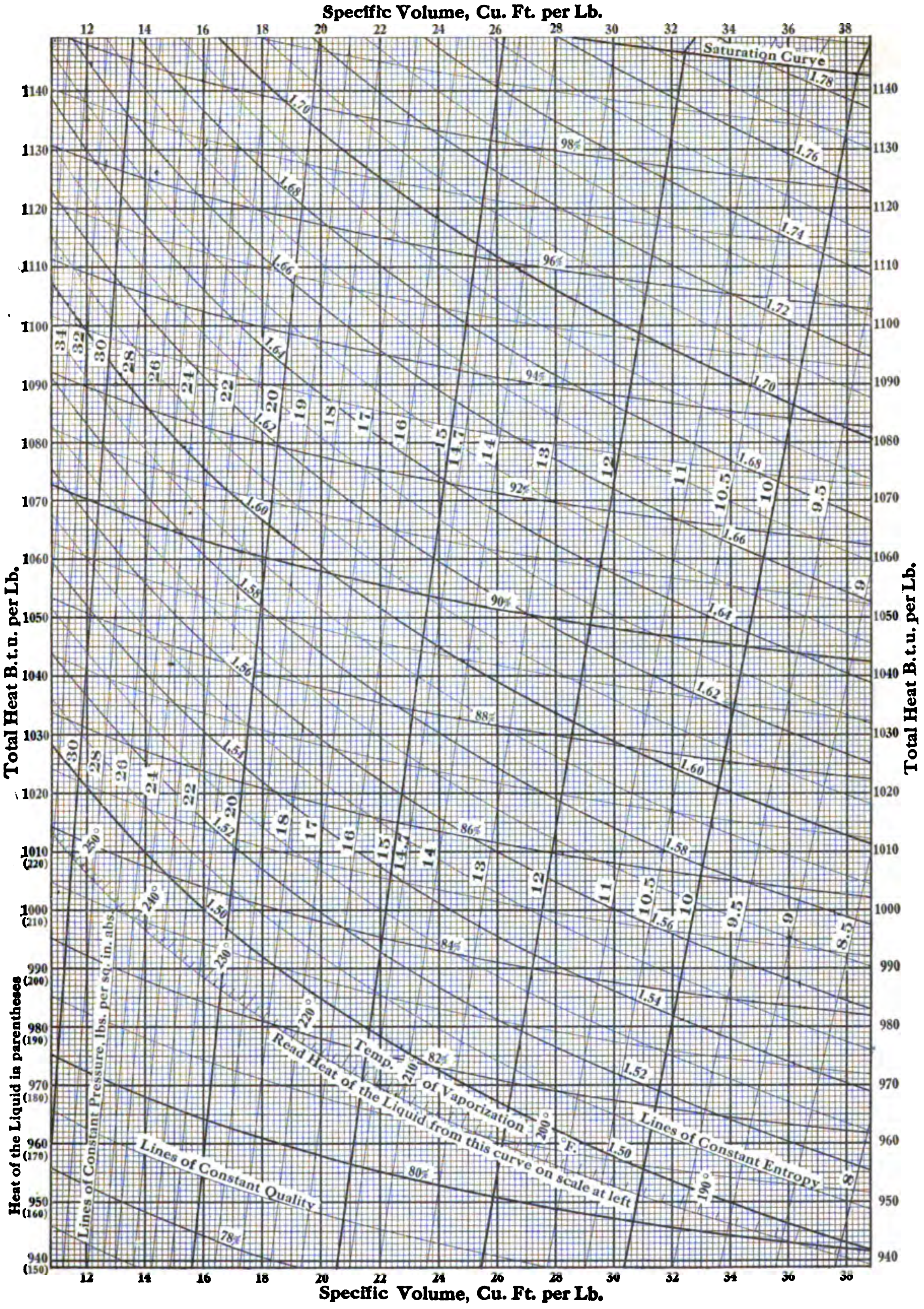

Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Lines of Constant Superheat-F.
Lines of Constant Entropy
Lines of Constant Pressure-lbs. per sq. in. abs.
Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.

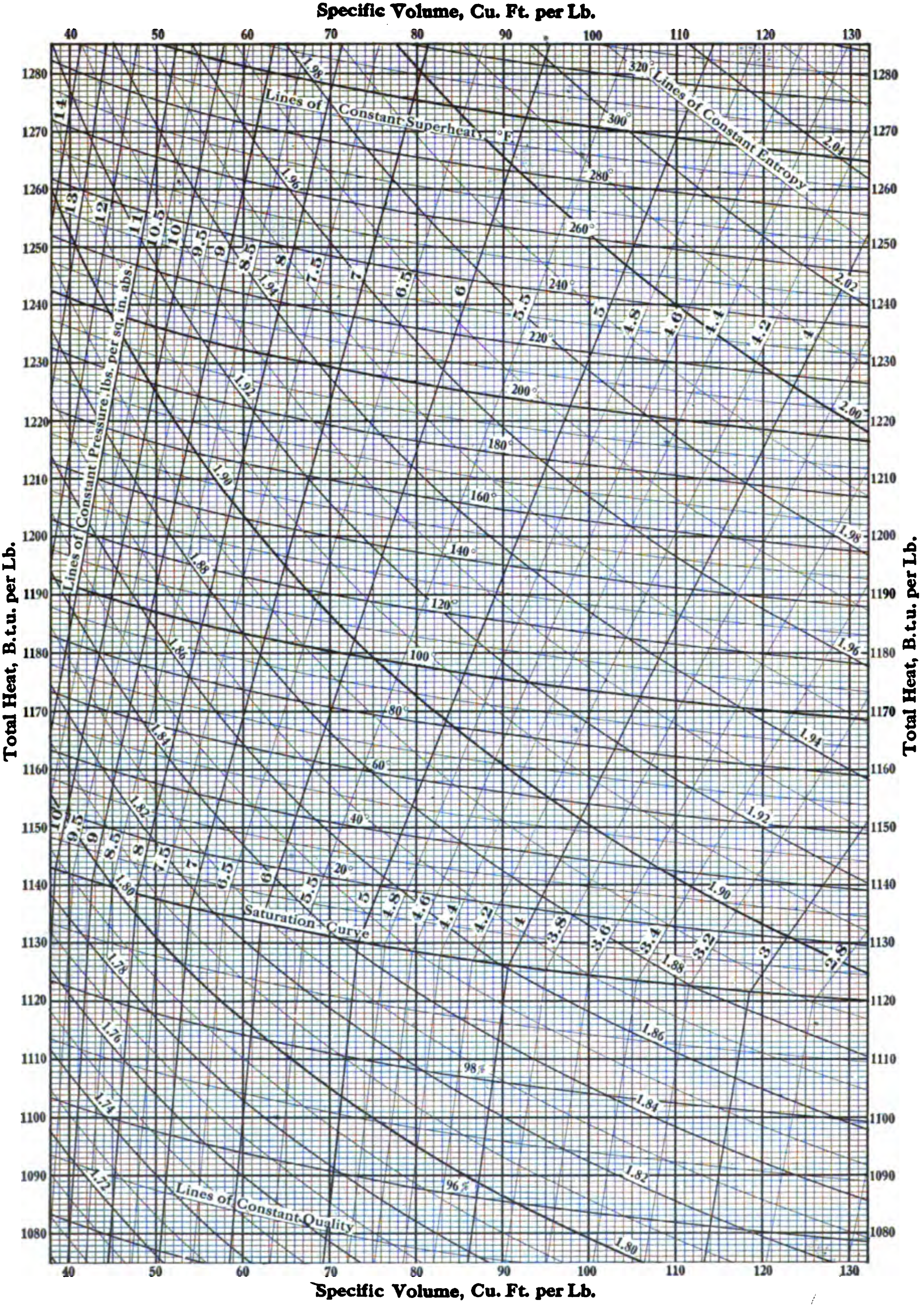

Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Heat of Liquid in parentheses
Lines of Constant Superheat °F.
Saturation Curve
Lines of Constant Entropy
Lines of Constant Pressure, lbs. per sq. in. abs.
Lines of Constant Quality
Temp. of Vaporization
Read Heat of the Liquid from this curve on scale at left
Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.

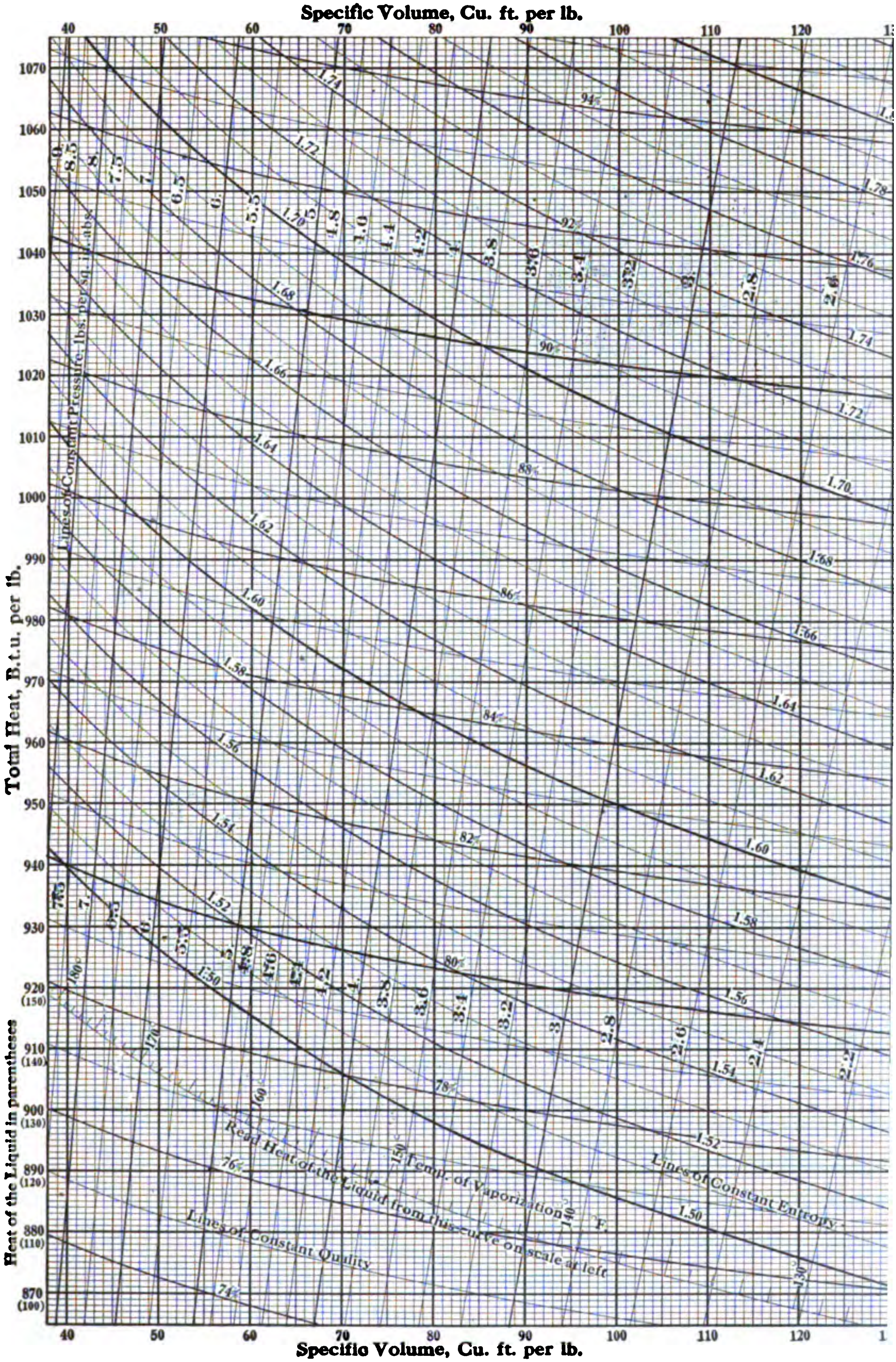

Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Lines of Constant Superheat °F
Lines of Constant Entropy
Lines of Constant Pressure, lbs. per sq. in. abs.
4
5
6
7
8
9
10
1240
1250
1260
1270
1280
1290
1300
1310
1320
1330
1340
1350
1360
1370
1380
1390
1400
1410
1420
1430
1440

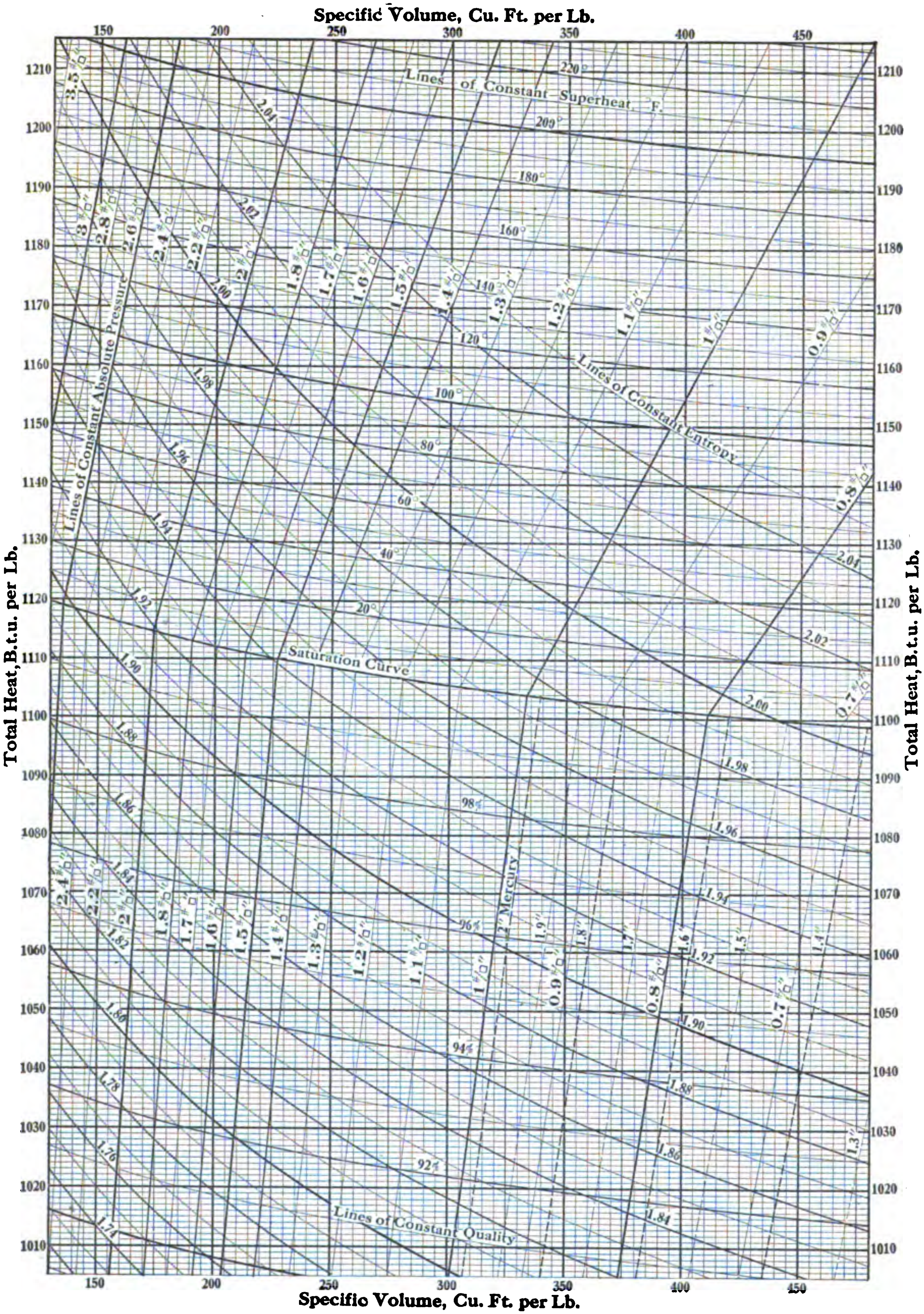

Specific Volume, Cu. ft. per lb.
Total Heat B.t.u. per lb.
Heat of the Liquid in parentheses
Lines of Constant Superheat °F.
Saturation Curve
Lines of Constant Pressure, lbs. per sq. in. abs.
Read Heat of the Liquid from this curve on scale at left.
Temp. of Vaporization, °F.
Lines of Constant Quality
98%
96%
94%
92%
90%
88%
86%
84%

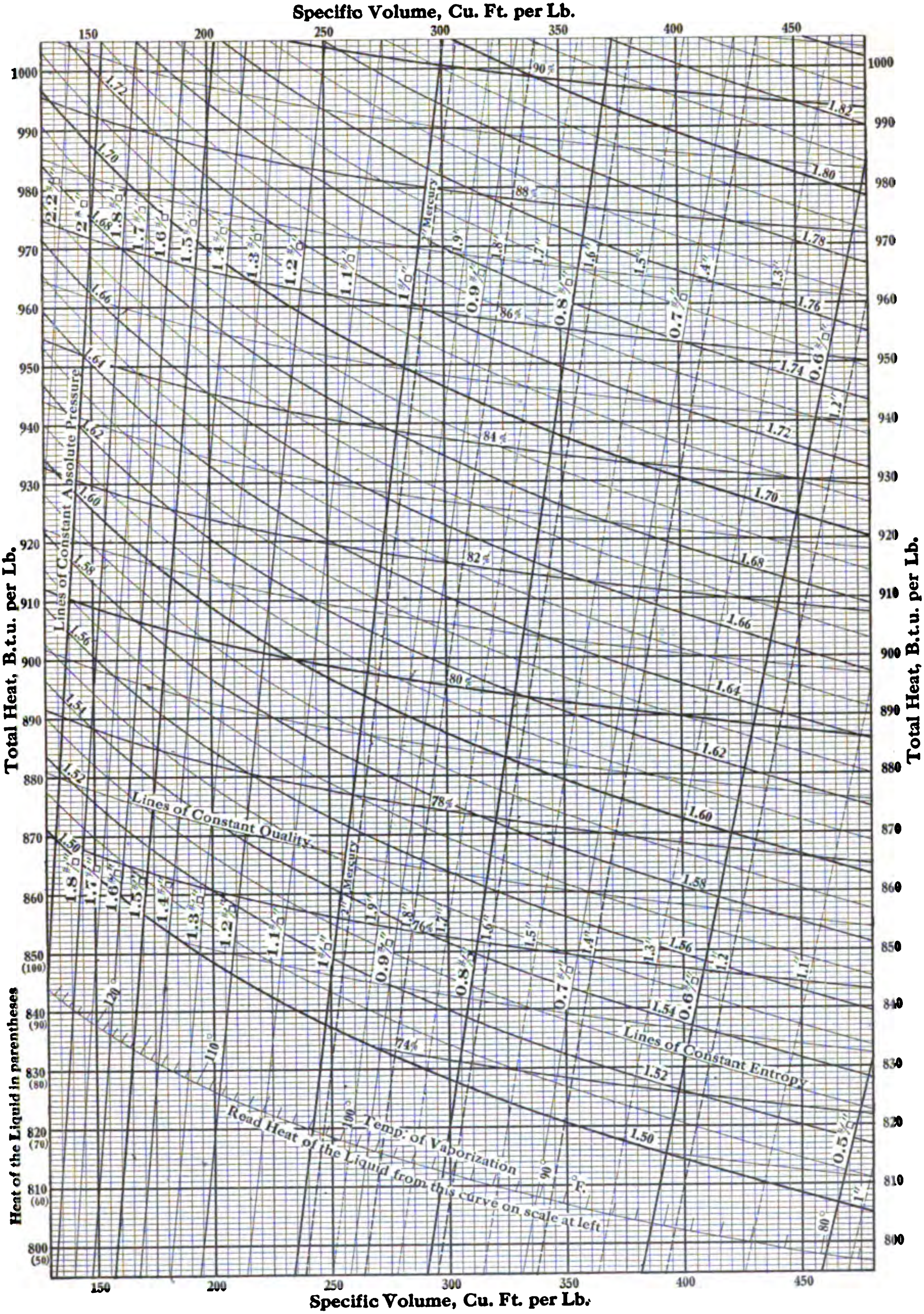

Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per lb.
Lines of Constant Superheat F.
Lines of Constant Entropy
Lines of Constant Pressure, lbs. per sq. in. abs.
Saturation Curve

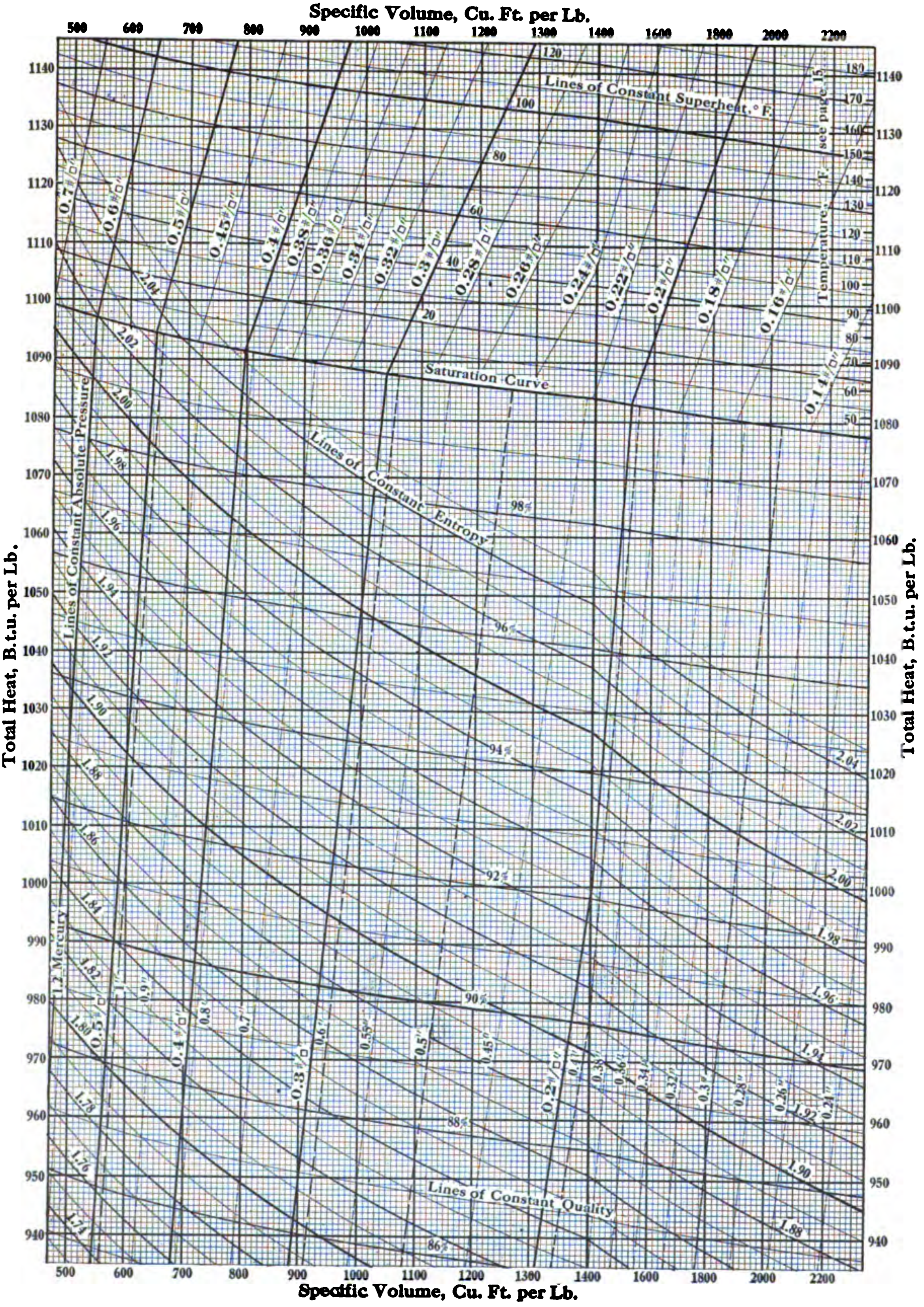

Specific Volume, Cu. Ft. per Lb.
Total Heat B.t.u. per Lb.
Heat of the Liquid in parentheses
Saturation Curve
Lines of Constant Pressure, lbs. per sq. in. abs.
Lines of Constant Quality
Lines of Constant Entropy
Temp. of Vaporization °F.
Read Heat of the Liquid from this curve on scale at left

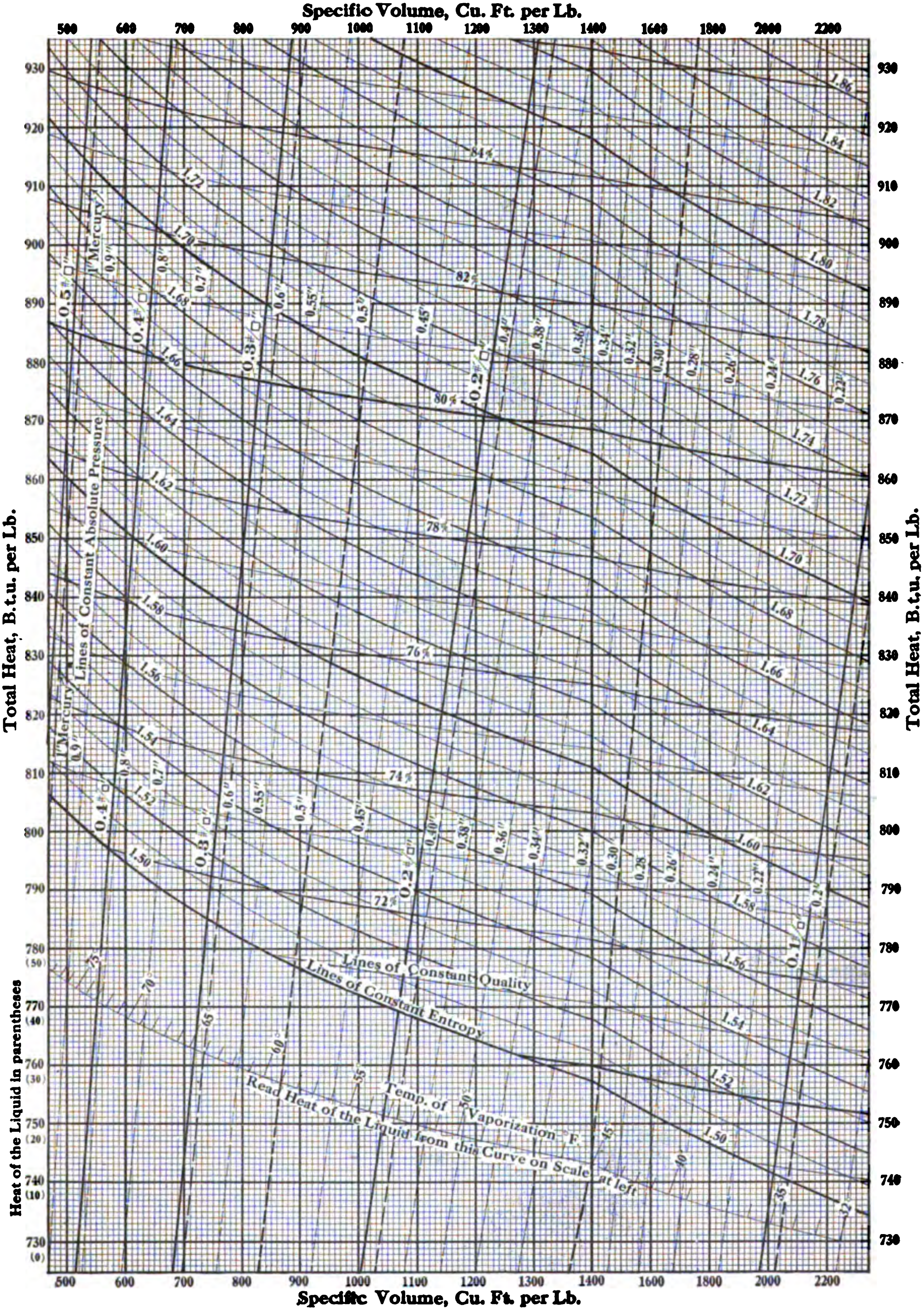

Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Lines of Constant Superheat, °F
Lines of Constant Entropy
Lines of Constant Pressure, lbs. per sq. in. abs.
Saturation Curve
Lines of Constant Quality

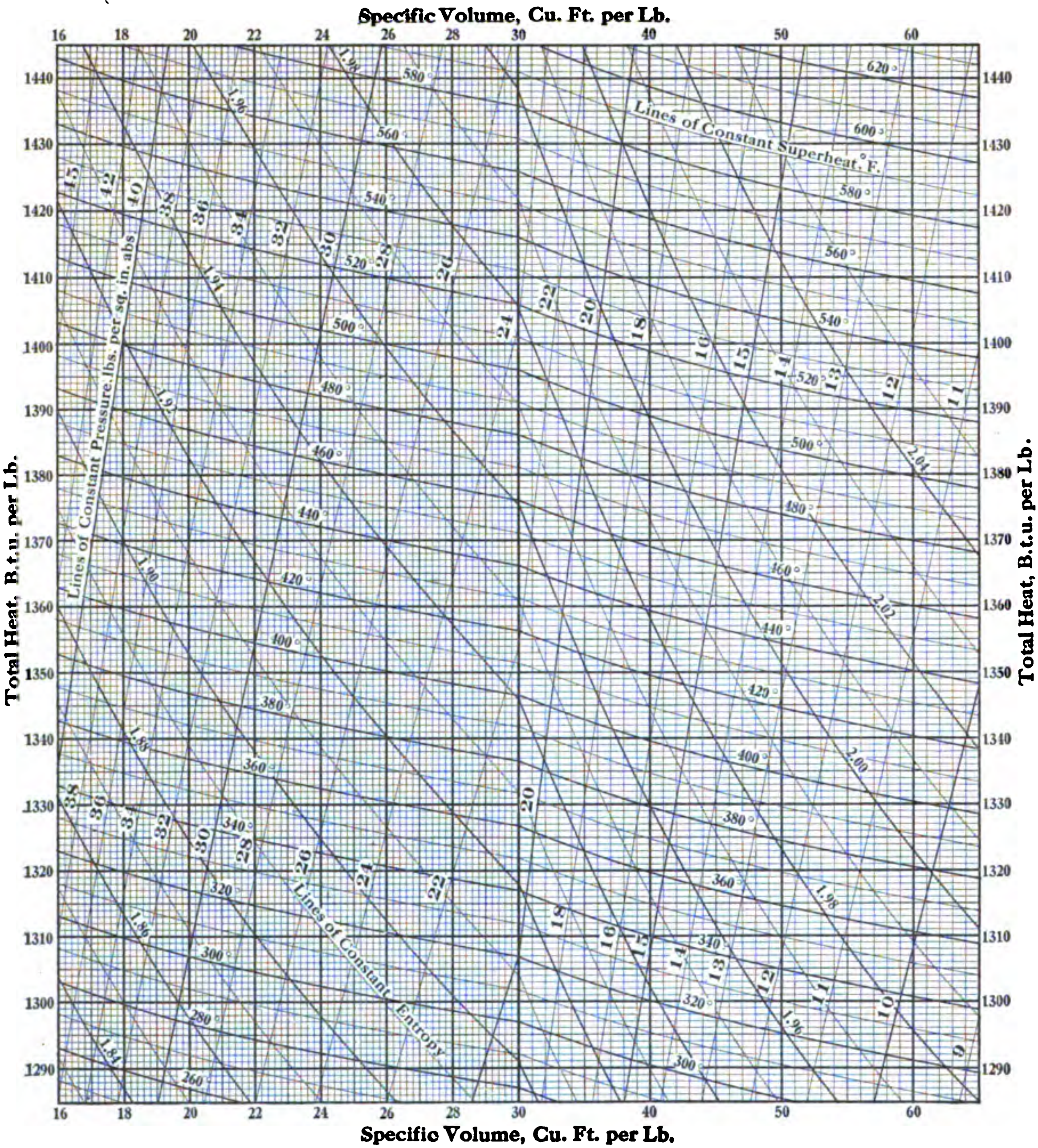

Specific Volume, Cu. ft. per lb.
Total Heat, B.t.u. per lb.
Heat of the Liquid in parentheses
Lines of Constant Pressure, lbs. per sq. in. abs.
Lines of Constant Entropy
Lines of Constant Quality
Temp. of Vaporization °F.
Read Heat of the Liquid from this curve on scale at left
Specific Volume, Cu. ft. per lb.


Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Lines of Constant Superheat, °F.
Lines of Constant Absolute Pressure
Lines of Constant Entropy
Saturation Curve
Lines of Constant Quality
2" Mercury


Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Heat of the Liquid in parentheses
Lines of Constant Absolute Pressure
Lines of Constant Quality
Lines of Constant Entropy
Temp. of Vaporization °F.
Read Heat of the Liquid from this curve on scale at left
Mercury

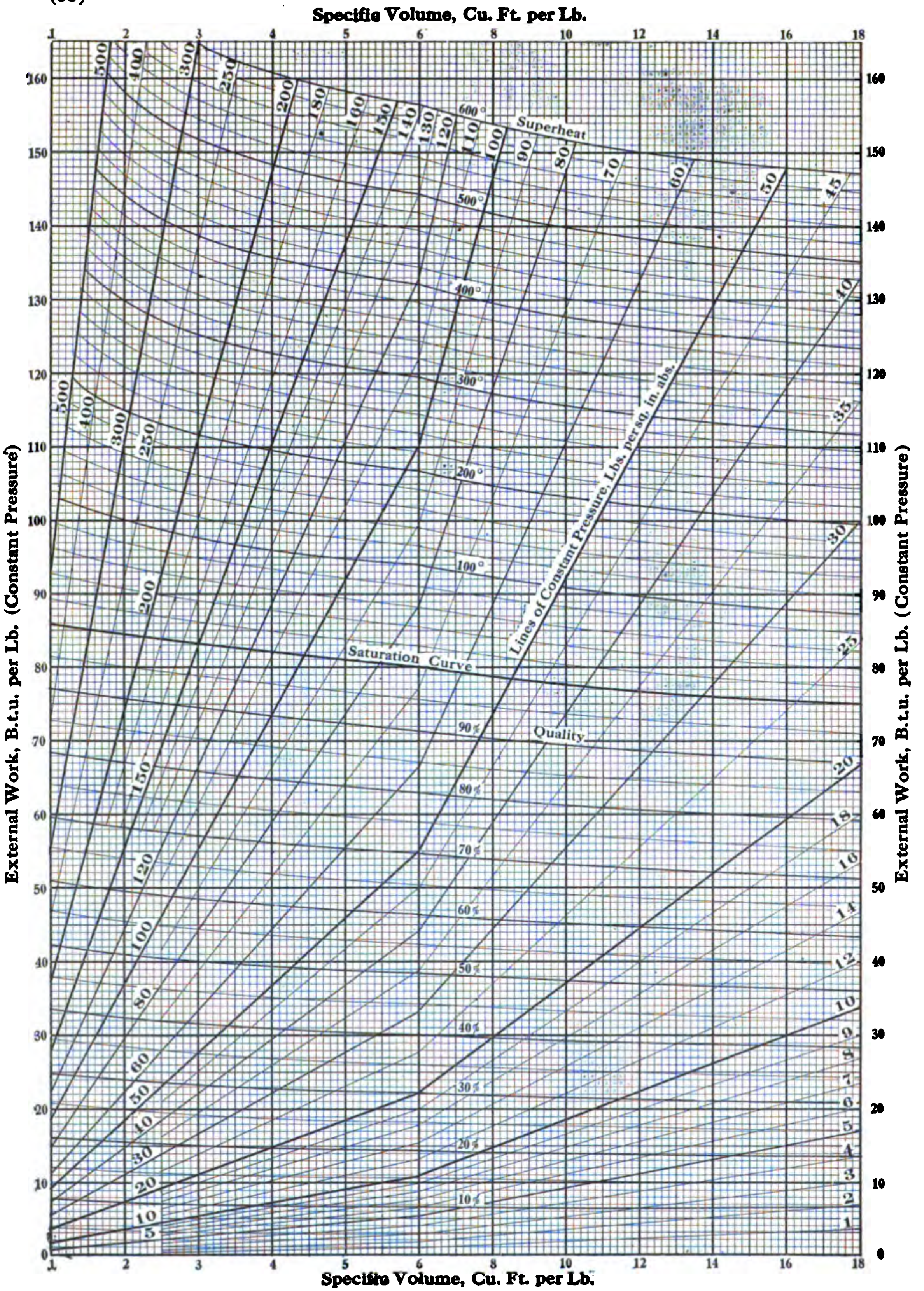

Specific Volume, Cu. Ft. per Lb.
Total Heat, B.t.u. per Lb.
Lines of Constant Superheat, °F.
Temperature, °F. see page 15
Saturation Curve
Lines of Constant Entropy
Lines of Constant Absolute Pressure
Lines of Constant Quality
1.2" Mercury


Specific Volume, Cu. Ft. per Lb.
500 600 700 800 900 1000 1100 1200 1300 1400 1600 1800 2000 2200
Total Heat, B.t.u. per Lb.
930 920 910 900 890 880 870 860 850 840 830 820 810 800 790 780 770 760 750 740 730
Heat of the Liquid in parentheses
(50) (40) (30) (20) (10) (0)
Lines of Constant Absolute Pressure
Lines of Constant Quality
Lines of Constant Entropy
Temp. of Vaporization °F.
Read Heat of the Liquid from this Curve on Scale at left
84% 82% 80% 78% 76% 74% 72%
1.86 1.84 1.82 1.80 1.78 1.76 1.74 1.72 1.70 1.68 1.66 1.64 1.62 1.60 1.58 1.56 1.54 1.52 1.50
1" Mercury
0.5#/□" 0.4#/□" 0.3#/□" 0.2#/□" 0.1#/□"
0.9" 0.8" 0.7" 0.6" 0.55" 0.5" 0.45" 0.4" 0.38" 0.36" 0.34" 0.32" 0.30" 0.28" 0.26" 0.24" 0.22" 0.2"
75° 70° 65° 60° 55° 50° 45° 40° 35° 32°



THIS PLATE IS TO SUPPLEMENT PLATES 3A AND 4A FOR THOSE EXCEPTIONAL CASES IN WHICH EXTREMELY HIGH SUPERHEAT IS USED FOR COMPARATIVELY LOW PRESSURES.

THE TWO FOLLOWING PLATES GIVE THE EXTERNAL WORK DONE DURING THE CONSTANT PRESSURE FORMATION OF STEAM FROM WATER AT 32° F.


Specific Volume, Cu. Ft. per Lb.
External Work, B.t.u. per Lb. (Constant Pressure)
Superheat
600°
500°
400°
300°
200°
100°
Saturation Curve
Quality
90%
80%
70%
60%
50%
40%
30%
20%
10%
Lines of Constant Pressure, Lbs. per sq. in. abs.

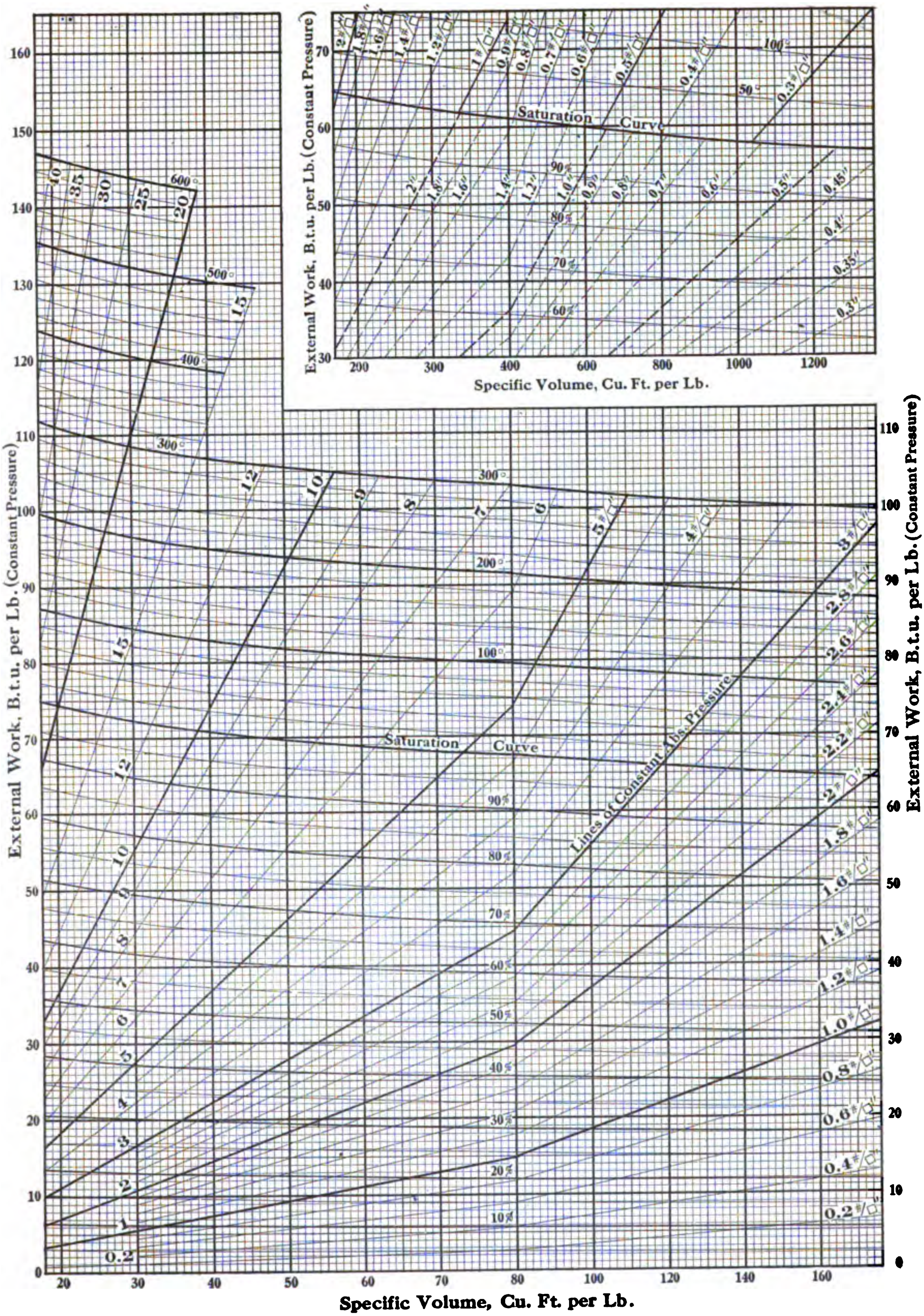

External Work, B.t.u. per Lb. (Constant Pressure)
Specific Volume, Cu. Ft. per Lb.
Saturation Curve
Lines of Constant Abs. Pressure
External Work, B.t.u. per Lb. (Constant Pressure)
Saturation Curve
Specific Volume, Cu. Ft. per Lb.

Plate 9A

Showing correction of Mercury Column due to Temperature, when scale is correct at the observed temperature

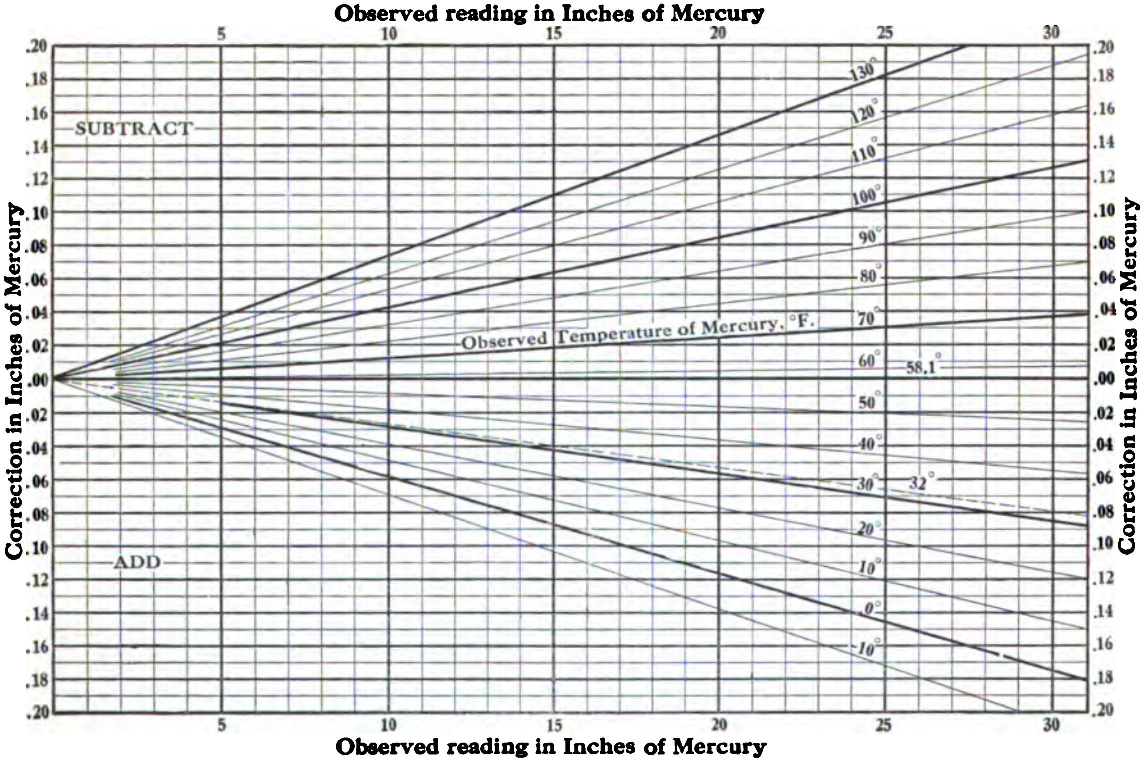


Plate 9B

Showing correction of Barometric Reading due to change in Elevation

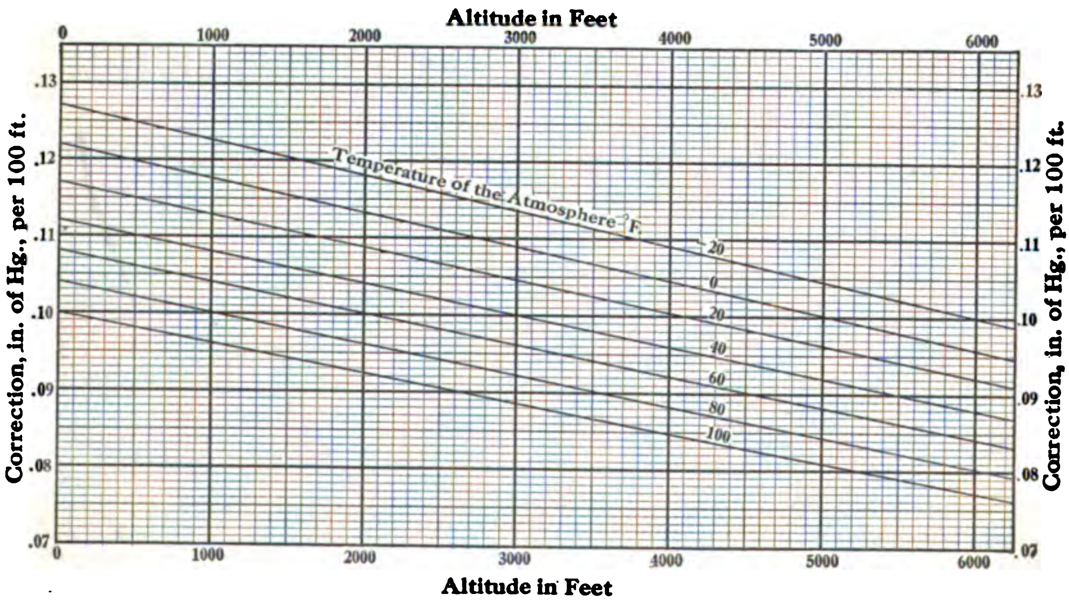


TABLE I

SHOWING CORRECTIONS TO REDUCE BAROMETRIC READINGS TO 45° LATITUDE *

For These Latitudes the Correction is to be Subtracted	Barometer Reading in Inches of Mercury									For These Latitudes the Correction is to be Added
	22	23	24	25	26	27	28	29	30	
0°	.059	.061	.064	.067	.069	.072	.074	.077	.080	90°
10°	.055	.058	.060	.063	.065	.068	.070	.073	.075	80°
20°	.045	.047	.049	.051	.053	.055	.057	.059	.061	70°
30°	.029	.031	.032	.033	.035	.036	.037	.039	.040	60°
40°	.010	.011	.011	.012	.012	.012	.013	.013	.014	50°
45°	.000	.000	.000	.000	.000	.000	.000	.000	.000	45°

* Abridged and rearranged from Table 101 of the Smithsonian Tables.

TABLE II

CORRECTION OF THE BAROMETER FOR CAPILLARITY †

Correction to be added in inches

Diameter of Tube in Inches	Height of Meniscus in Inches							
	.01	.02	.03	.04	.05	.06	.07	.08
.15	.024	.047	.069	.092	.116	....	....	....
.20	.011	.022	.033	.045	.059	.079	....	....
.25	.006	.012	.019	.028	.037	.047	.059	....
.30	.004	.008	.013	.018	.023	.029	.035	.042
.35	....	.005	.008	.012	.015	.019	.022	.027
.40	....	.004	.006	.008	.010	.012	.014	.016
.45	....	....	.003	.005	.007	.008	.010	.012
.50	....	....	.002	.004	.005	.006	.006	.007
.55	....	....	.001	.002	.003	.004	.005	.005

† From Table 103 Smithsonian Tables, modified by giving the correction only to the nearest thousandth of an inch.

TABLE III

DENSITY OF MERCURY ‡

Temperature °F.	Pounds per Cubic Inch	Temperature °F.	Pounds per Cubic Inch
0	.4928	**58.1**	**.4899**
10	.4923	60	.4898
20	.4918	70	.4893
30	.4913	80	.4888
32	**.4912**	90	.4883
40	.4907	100	.4878
50	.4903	110	.4873

‡ See page 22.

TABLE IV
SHOWING THE THEORETICAL VELOCITIES ATTAINED BY STEAM EXPANDING ADIABATICALLY IN A FRICTIONLESS NOZZLE

The velocities are given in feet per second for each B. t. u. up to 599.

Available Energy B. t. u. per lb.	0	1	2	3	4	5	6	7	8	9
0	0	224	316	387	447	500	548	592	633	671
10	707	742	775	806	837	866	895	922	949	975
20	1001	1026	1050	1073	1097	1120	1141	1163	1184	1205
30	1226	1246	1266	1285	1304	1323	1342	1361	1379	1397
40	1415	1433	1450	1467	1484	1501	1517	1533	1550	1566
50	1582	1598	1613	1628	1643	1658	1673	1688	1703	1718
60	1732	1747	1761	1775	1789	1803	1817	1831	1844	1858
70	1872	1885	1898	1911	1924	1937	1950	1963	1976	1988
80	2000	2013	2026	2038	2050	2062	2074	2086	2098	2110
90	2122	2134	2146	2158	2169	2180	2191	2202	2214	2226
100	2237	2248	2259	2270	2281	2292	2303	2314	2325	2336
110	2346	2356	2367	2378	2389	2399	2409	2419	2430	2440
120	2450	2460	2470	2480	2490	2500	2511	2521	2531	2540
130	2550	2560	2570	2580	2590	2600	2609	2619	2628	2637
140	2647	2657	2666	2675	2684	2694	2703	2712	2721	2730
150	2740	2749	2758	2767	2776	2785	2794	2803	2812	2821
160	2830	2839	2848	2857	2866	2874	2882	2891	2900	2908
170	2917	2925	2934	2942	2951	2960	2968	2976	2984	2993
180	3001	3010	3018	3026	3034	3042	3050	3059	3067	3075
190	3083	3091	3100	3108	3116	3124	3132	3140	3148	3156
200	3164	3172	3180	3188	3196	3204	3211	3219	3227	3234
210	3241	3249	3257	3265	3273	3280	3288	3296	3303	3310
220	3318	3325	3332	3340	3348	3355	3363	3370	3377	3384
230	3392	3400	3407	3414	3422	3430	3437	3444	3451	3458
240	3465	3473	3480	3487	3494	3501	3508	3516	3523	3530
250	3537	3544	3551	3558	3565	3572	3579	3586	3593	3600
260	3607	3614	3620	3627	3634	3641	3648	3655	3662	3669
270	3676	3683	3689	3696	3703	3710	3717	3723	3730	3737
280	3743	3750	3757	3763	3770	3777	3783	3790	3796	3803
290	3810	3817	3823	3829	3835	3842	3849	3855	3861	3868

TABLE IV

SHOWING THE THEORETICAL VELOCITIES ATTAINED BY STEAM EXPANDING ADIABATICALLY IN A FRICTIONLESS NOZZLE

The velocities are given in feet per second for each B. t. u. up to 599.

Available Energy B. t. u. per lb.	0	1	2	3	4	5	6	7	8	9
300	3874	3881	3888	3894	3900	3907	3913	3920	3926	3932
310	3939	3946	3952	3958	3964	3970	3976	3982	3989	3995
320	4002	4008	4014	4020	4027	4033	4039	4045	4051	4058
330	4063	4070	4076	4082	4088	4094	4100	4107	4113	4119
340	4125	4131	4137	4143	4149	4155	4161	4167	4173	4179
350	4185	4191	4197	4203	4209	4215	4221	4227	4233	4239
360	4245	4251	4257	4263	4268	4274	4280	4286	4291	4297
370	4302	4308	4314	4320	4326	4332	4338	4344	4350	4356
380	4361	4367	4372	4378	4383	4389	4395	4401	4407	4413
390	4418	4424	4430	4435	4440	4446	4451	4457	4462	4468
400	4473	4479	4485	4490	4496	4502	4508	4513	4519	4524
410	4530	4536	4541	4547	4552	4558	4563	4569	4574	4580
420	4585	4590	4596	4601	4607	4612	4617	4623	4628	4634
430	4639	4644	4650	4655	4661	4666	4671	4677	4682	4688
440	4693	4698	4703	4709	4714	4719	4724	4729	4735	4740
450	4745	4750	4755	4761	4766	4771	4776	4781	4787	4792
460	4797	4802	4808	4813	4818	4823	4828	4833	4839	4844
470	4849	4854	4859	4865	4871	4875	4880	4885	4891	4896
480	4901	4906	4911	4917	4922	4927	4932	4937	4942	4947
490	4952	4957	4962	4967	4972	4977	4982	4987	4992	4997
500	5002	5007	5012	5017	5022	5027	5032	5037	5042	5047
510	5052	5057	5062	5067	5072	5077	5082	5087	5091	5096
520	5101	5106	5111	5116	5121	5126	5131	5136	5140	5145
530	5150	5155	5160	5164	5169	5174	5179	5184	5188	5193
540	5198	5203	5208	5212	5217	5222	5227	5232	5236	5241
550	5246	5251	5256	5260	5265	5270	5275	5280	5284	5289
560	5294	5299	5303	5308	5312	5317	5322	5327	5331	5336
570	5341	5346	5350	5355	5359	5364	5369	5373	5378	5382
580	5387	5392	5397	5401	5406	5411	5416	5420	5425	5429
590	5434	5439	5443	5448	5452	5457	5461	5466	5470	5475

PROBLEMS

1. Find the total heat, volume, entropy, and temperature of a pound of steam, having an absolute pressure of 75 pounds per square inch, and superheated 280° F.

Solution: Referring to the index chart, it will be seen that for this pressure Plates 2a and 2b should be used.

From Plate 2a, we read directly:

Total heat = 1320.7 B. t. u.
Volume = 8.28 cu. ft.
Entropy = 1.78

From Plate 2b, we read from the temperature of vaporization curve near the bottom of the Plate:

Temp. of vaporization for 75 lbs. per sq. in. abs. = 307.6° F.

Hence, for 280° superheat,

Temperature = 280 + 307.6 = 587.6° F.

2. Find the total heat, volume, entropy, temperature, and heat of the liquid of a pound of steam having an absolute pressure of 180 pounds per square inch and a quality of 98%.

Solution: From Plate 1b, we read directly:

Total heat = 1179.3 B. t. u.
Volume = 2.48 cu. ft.
Entropy = 1.534
Temperature = 373.1° F.
Heat of the liquid = 345.5 B. t. u.

3. Eight pounds of steam are confined in a space of 800 cubic feet. If the quality of this steam is 92%, find its pressure and temperature.

Solution:

$$\text{Specific volume} = \frac{800}{8} = 100 \text{ cu. ft. per lb.}$$

Turning to Plate 4b and following up this specific-volume line, 100 cu. ft., until we intersect the quality of 92%, we read:

Pressure = 3.3 pounds per sq. in. absolute

By running down this pressure line, which is approximately halfway between the pressure lines marked 3.2 and 3.4 pounds per square inch, until we intersect the curve graduated to give temperatures of vaporization, we read:

Temperature = 145.5° F.

4. The barometer reads 30.05 inches of mercury at a temperature of 91° F. The mercury column attached to the condenser reads 28.6 inches at a temperature of 110° F. Find the absolute pressure in inches of mercury in the condenser at the standard temperature of 58.1° F.*

Solution: By referring to Plate 9a, we read:

For 30.05″ at 91° correction is − .10″
Hence barometer reading at 58.1° F. = 30.05 − .10 = 29.95″
Likewise the correction for the vacuum reading of 28.6″ at 110° F. is seen to be − 0.15″.
Therefore vacuum reading at 58.1° = 28.6 − .15 = 28.45″ and
absolute pressure = 29.95 − 28.45 = 1.5″ Hg. at 58.1°.

5. One pound of steam expands at constant entropy of 1.69 until condenser pressure is attained. The manometer on this condenser reads 29.50 inches of mercury at a temperature of 92° F., and the barometer reading is 30.04 inches of mercury at 38° F. Find the total heat, volume, quality, and temperature of this steam.

* This is the temperature at which 30 inches of mercury are equal to a standard atmosphere, which is defined as being equal to 29.921 inches of mercury at 32° F. See page 18 for further discussion of this subject.

Solution: It is first necessary to obtain the absolute condenser pressure, measured in inches of mercury at the temperature 58.1°, which is the temperature for which the mercury readings are given on the steam chart.

From Plate 9a, we read:

Barometer, 30.04″ at 38° = 30.04 + .06 or 30.10″ at 58.1° F.

Vacuum, 29.50″ at 92° = 29.50 − .10 or 29.40″ at 58.1° F.

Therefore absolute pressure in condenser must be 30.10 − 29.40 = 0.7 inches of mercury at 58.1° F.

Now, referring to Plate 6b, we read for an entropy of 1.69 and a pressure of 0.7 inches of mercury, the following values:

Total heat = 891 B. t. u.
Volume = 745 cu. ft.
Quality = 81.1%
Temperature = 68.6° F.

6. How many pounds of steam are contained in a steam pipe which is 12 inches in diameter and 200 feet long, if the average pressure in this pipe is 100 pounds per square inch absolute, and the average temperature is 433.5° F.?

Solution: By referring to Plate 1b, we find that the temperature of vaporization of steam having an absolute pressure of 150 pounds per square inch is 358.5° F.

Hence degrees of superheat = 433.5 − 358.5 = 75° F. Then from the same plate, for a pressure of 150 pounds per square inch and 75° superheat, we read:

Specific volume = 3.4 cu. ft. per pound.

Hence weight of steam in this pipe must be

$$\frac{\text{volume of pipe}}{\text{specific volume of steam}} = \frac{1 \times .7854 \times 200}{3.4} = 46.2 \text{ pounds.}$$

7. Find the number of B. t. u. which are required to do the external work during the formation of a pound of steam at a con-

stant pressure of 60 pounds per square inch absolute from water at 32° F. until it is superheated 210° F.

Solution: Turning to Plate 8a, at the intersection of the 60-pound line and the 210° superheat line, we read:

External work done during this constant pressure change
= 105 B. t. u.

8. Find the intrinsic heat of the steam in the final state given in the previous problem.

Solution: For an absolute pressure of 60 pounds per square inch and 210° superheat, from Plate 2a we read:

Total heat = 1281 B. t. u.

Then, since the intrinsic heat is merely that heat contained within the steam itself, we may obtain it easily by subtracting from the total heat the external work done by the steam during its constant pressure formation from water at 32° F. This external work, from previous problems, is 105 B. t. u.

Hence, for this case,

Intrinsic heat = 1281 − 105 = 1176 B. t. u.

9. If a pound of dry saturated steam is heated at a constant pressure of 200 pounds per square inch absolute, until its total heat is increased by 10%, find how many degrees it has been superheated and the per cent. increase in its volume.

Solution: From Plate 1b, we find:

Initial total heat = 1198.2 B. t. u.

Initial volume = 2.29 cu. ft.

Hence final total heat = 1198.2 x 1.10 = 1318 B. t. u. From Plate 1a, for this total heat and a pressure of 200, we read:

Final volume = 3.11 cu. ft.

Final superheat = 221°.

Therefore per cent. increase in volume $= \dfrac{3.11 - 2.29}{2.29} = 35.8\%$.

10. Steam having an absolute pressure of 80 pounds per square inch and a quality of 98% is passed through a superheater on its way to an engine. If the steam leaving the superheater has a temperature of 450° F., find:

(a) Degrees of superheat,
(b) Heat added per pound by superheater,
(c) Specific volume of the superheated steam,
(d) What per cent. of the heat added by this superheater was required to complete the vaporization.

Solution: For steam having an absolute pressure of 80 pounds per square inch, we find, from Plate 2b:

Temperature of vaporization = 312° F.

Hence

(a) Degrees of superheat = 450 − 312 = 138° F.

Also from this same plate, for the given pressure and quality, we observe:

Initial total heat = 1164 B. t. u.

Then, by running along the 80-pound pressure line until it intersects the 138° superheat line, we find, from Plate 2a:

Final total heat = 1253 B. t. u.

Hence

(b) Heat added by superheater = 1253 − 1164 = 89 B. t. u.

Also from Plate 2a, we may read:

(c) Specific volume = 6.65 cu. ft. per pound.

For the given pressure we may read, from Plate 2b:

(d) Total heat of dry saturated steam = 1182.3 B. t. u.

Hence the heat required to complete the vaporization of the wet steam is

1182.3 − 1164 = 18.3 B. t. u.

Therefore the portion of the heat which was added by the superheater in order to complete the vaporization is

$$\frac{18.3}{89} = 20.6\%$$

11. If one pound of steam expands adiabatically from an absolute pressure of 100 pounds per square inch and 145° superheat, to an absolute pressure of 1.4 inches of mercury, find:

(a) Final quality.
(b) Final volume.
(c) Ratio of expansion.
(d) Work done.

Solution: From Plate 2a, for the 100 pound line and 145° superheat, we read:

Initial total heat = 1262.3 B. t. u.
Initial entropy = 1.69
Initial volume = 5.45 cu. ft.

Then turning to Plate 5b, and running along the entropy line 1.69 until it intersects the pressure line 1.4 inches of mercury, we read:

Final total heat = 925 B. t. u.
(a) Final quality = 83.4%
(b) Final volume = 397 cu. ft.

From these results we have:

$$\text{(c) Ratio of expansion} = \frac{\text{Final volume}}{\text{Initial volume}} = \frac{397}{5.45} = 72.8$$

In order to obtain the work done during this expansion, it is necessary to obtain the intrinsic heats for the initial and final states, since the work done during adiabatic expansion is equal to the loss of intrinsic heat. The intrinsic heat for any state may be obtained by subtracting the external work, as obtained from Plate 8, from the total heat.

For the initial pressure of 100 pounds and superheat of 145°, from Plate 8a, we read:

The constant pressure external work = 100.3 B. t. u.
Hence

Initial intrinsic heat = 1262.3−100.3 = 1162 B. t. u.

For the final pressure of 1.4″ Hg. and a quality of 83.4%, we read from the upper right-hand corner of Plate 8b:

The constant pressure external work = 50 B. t. u.

Hence

Final intrinsic heat = 925 − 50 = 875 B. t. u.

(d) Work done during adiabatic expansion = 1162 − 875
= 287 B. t. u. or 223300 ft. lbs.

12. Plot to scale, on a pressure volume diagram, the line representing the adiabatic expansion for the previous problem.

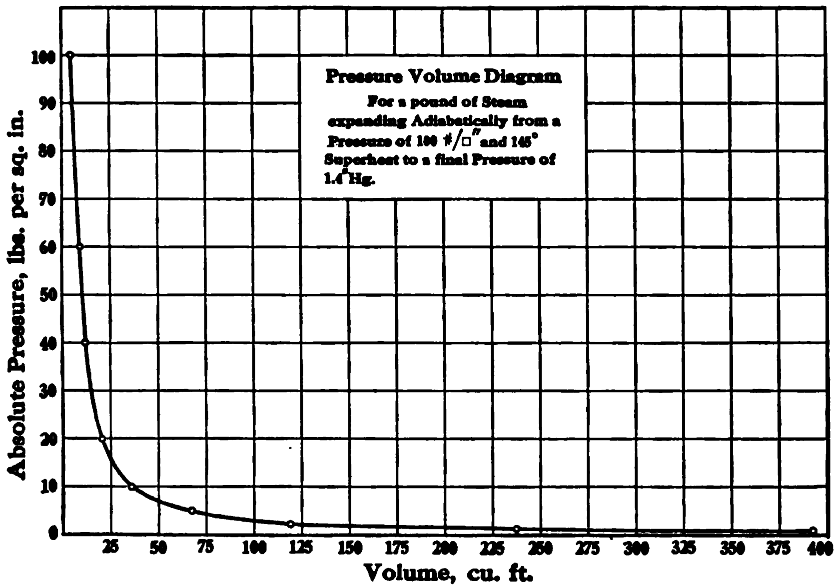


Fig. 3.

Solution: This curve is easily obtained by reading the volumes determined by the constant entropy line, 1.69, cutting the various pressure lines, assumed as desired. Thus:

Plate used	Pressure	Volume
2a	100	5.45
2b	60	8.00
2b	40	10.85

Plate used	Pressure	Volume
3b	20	19.5
3b	10	35.9
4b	5	66.4
4b	2.6	119.3
5b	1.2	237.
5b	1.4″ Hg.	397.

13. A closed metallic tank having a cubical content of 30 cubic feet contains 3 pounds of steam having an absolute pressure of 75 pounds per square inch.

If heat is now abstracted from this tank by pouring cold water over it until the pressure of the steam within has fallen to 40 pounds per square inch absolute, find:

(a) The initial superheat.
(b) The final quality.
(c) The heat abstracted.

Solution: Specific Volume $= \frac{30}{3} =$ 10 cu. ft. per lb.

Hence by turning to Plate 2a, and running along the 75-pound line until it intersects the 10 cubic feet specific-volume line, we find:

(a) Initial superheat = 490° F.
Initial total heat = 1421 B. t. u. per lb.

(b) This must be a constant volume abstraction of heat since the steam is confined in a closed tank. Hence for a specific volume of 10 cubic feet per pound and an absolute pressure of 40 pounds per square inch we may read from Plate 2b:

Final quality = 95.3%
Final total heat = 1126 B. t. u. per lb.

(c) The heat abstracted during any constant-volume process is equal to the loss of intrinsic heat, since no work is done during such a process.

The intrinsic heat for any state may be obtained by subtracting the external work, as obtained from Plate 8, from the total heat.

At the intersection of the 10 cubic feet specific-volume line and the 490° superheat line on Plate 8a, we read:

The constant pressure external work = 138.5 B. t. u. per lb.
Hence
Initial intrinsic heat = 1421 − 138.5 = 1282.5 B. t. u. per lb.

From the same plate, at the intersection of the 40-pound pressure line and the 10 cubic feet specific-volume line, we read:

The constant pressure external work = 73.8 B. t. u. per lb.
Hence final intrinsic energy must be

1126 − 73.8 = 1052.2 B. t. u. per lb.

Therefore for the 3 pounds of steam cooled at constant volume, we have:

Heat abstracted = 3 [1282.5 − 1052.2]
= 3 [230.3] = 690.9 B. t. u.

14. A pound of steam having a pressure of 62 pounds per square inch absolute and a temperature of 340° F. is heated isothermally until its pressure becomes 20 pounds per square inch absolute.

Find:

(a) Initial and final superheats.
(b) Initial and final entropies.
(c) Initial and final total heats.
(d) Initial and final volumes.
(e) Initial and final intrinsic heats.
(f) Heat required to effect the change.
(g) Work done during this expansion.

Solution: From Plates 2b and 3b respectively, we may read:

Temperature of vaporization for 62 lbs. = 295° F.
Temperature of vaporization for 20 lbs. = 228° F.

Therefore

(a) Initial superheat = 340 − 295 = 45° F.
Final superheat = 340 − 228 = 112° F.

Then from Plate 2b, for the pressure of 62 pounds and 45° superheat we may read:

(b_1) Initial entropy = 1.67
(c_1) Initial total heat = 1201 B. t. u.
(d_1) Initial volume = 7.46 cu. ft.

Also from Plate 3a, for the pressure of 20 pounds and 112° superheat we may read:

(b_2) Final entropy = 1.803
(c_2) Final total heat = 1209.3 B. t. u.
(d_2) Final volume = 23.7 cu. ft.

From Plate 8a, for the pressure of 62 pounds and the superheat of 45° we read:

The constant pressure external work = 85 B. t. u.

From Plate 8b, for the pressure of 20 pounds and the superheat of 112° we read:

The constant pressure external work = 87.3 B. t. u.

Then we may obtain:

(e) Initial intrinsic heat = 1201 − 85 = 1116 B. t. u.
Final intrinsic heat = 1209.3 − 87.3 = 1122 B. t. u.

The heat required to go along a constant temperature line is equal to the change in entropy multiplied by the absolute temperature.

Hence for this case

(f) Heat required = (1.803 − 1.67) (340 + 460)
= 106.4 B. t. u. given to the steam.

Representing the initial and final states by the subscripts 1 and 2, we then have, from the fundamental equation of thermodynamics:

$$\text{(g)} \quad \left.\begin{matrix}\text{Work}\\ \text{done}\end{matrix}\right]_1^2 = \left.\begin{matrix}\text{Heat}\\ \text{supplied}\end{matrix}\right]_1^2 - \left[\begin{matrix}\text{Gain in}\\ \text{intrinsic heat}\end{matrix}\right]_1^2$$
$$= 106.4 - [1122 - 1116]$$
$$= 106.4 - 6$$
$$= 100.4 \text{ B. t. u.}$$

NOTE.—This problem is of much more importance as a drill in thermodynamics than as a practical question; because it is very difficult to arrange the necessary apparatus to permit steam to expand isothermally while in the superheated field.

Those who think that the heat required to effect this change should be equal to the difference in total heats are referred to the meaning of that term as given in the introduction.

15. The pressure in a steam pipe is 105 pounds per square inch absolute, and in a throttling calorimeter, which is connected to this pipe, the pressure is 15 pounds per square inch absolute. If the temperature in the calorimeter is 238° F., find the quality of the steam.

Solution: From Plate 3b, for a pressure of 15 pounds we read:

$$\text{Temperature of vaporization} = 213° \text{ F.}$$

Hence

$$\text{Superheat in calorimeter} = 238 - 213 = 25° \text{ F.}$$

Then from the opposite page, Plate 3a, for a pressure of 15 pounds and 25° superheat we read:

$$\text{Total heat} = 1163 \text{ B. t. u.}$$

The throttling in a calorimeter takes place without any appreciable loss of heat by conduction or radiation, so the total heat remains constant.

Hence by running along this total heat line of 1163 B. t. u. until we intersect the 105-pound pressure line, we may read, from Plate 2b:

$$\text{Quality} = 97.3\%$$

16. If the temperature in the calorimeter of the previous problem had been 276° F., and the pressure in the calorimeter had been 16 pounds per square inch absolute, find the quality of the steam in the main having a pressure of 187 pounds per square inch absolute.

Solution: Proceeding as before, from Plate 3b we find the superheat in the calorimeter to be

$$276 - 216.3 = 59.7° \text{ F.}$$

Then, from Plate 3a, for the 16 pound line and 59.7° superheat we read:

$$\text{Total heat} = 1180 \text{ B. t. u.}$$

From Plate 1b, at the intersection of the 1180-total-heat line

and the 187-pound-pressure line, which is readily found by the eye between the 185- and 190-pound lines, we read:

Quality = 98%

17. The pressure in the seventh stage of a twelve-stage turbine is 10.5 pounds per square inch absolute, and its quality is estimated to be 94%. Supposing that a fair sample of the steam in this stage might be obtained, what pressure would have to be maintained in a throttling calorimeter in order that it might be used to determine this quality, provided that there shall be at least 10° superheat in the calorimeter?

Solution: From Plate 3b, for an absolute pressure of 10.5 pounds and a quality of 94% we read:

Total heat = 1085 B. t. u.

Then, running along this total-heat line until we intersect the 10° superheat line, we read from Plate 6a:

Pressure in calorimeter = 0.16 lbs. per sq. in. abs.

NOTE.—This extremely low pressure required in the calorimeter is very difficult to obtain and the scheme is not therefore a very practical one.

18. If steam having a quality of 99% is generated in an automobile boiler at a pressure of 400 pounds per square inch absolute, and is then throttled down to an absolute pressure of 100 pounds per square inch, find:

(a) Final superheat.
(b) Drop in temperature due to throttling.
(c) Increase in volume due to throttling.

Solution: From Plate 1b, for the pressure of 400 pounds and the quality of 99% we read:

Initial total heat = 1200 B. t. u. per pound
Initial temperature = 445° F. (almost)
Initial specific volume = 1.16 cubic feet per pound

Then from Plate 2b, running along the 1200-total-heat line until we intersect the 100-pound line, we read:

(a) Final superheat = 24° F.
Final specific volume = 4.6 cubic feet per pound
Temperature of vaporization for 100 pounds = 328° (almost)
Therefore final temperature = 328 + 24 = 352° F.

(b) The drop in temperature due to throttling then is

$$445 - 352 = 93° \text{ F.}$$

(c) The increase in specific volume due to throttling is

$$4.6 - 1.16 = 3.44 \text{ cu. ft. per lb.}$$

19. The test on a boiler gave the following data:

Absolute pressure, pounds per square inch	185
Superheat of steam, °F	125
Temperature of feed water, °F	200
Water evaporated per pound of coal, pounds	9.8
Calorific value of the coal, B. t. u., per pound	14300

Find the efficiency of the boiler and furnace combined.

Solution: For any temperature in the neighborhood of 200° F. the heat of the liquid may be accurately obtained by subtracting 32 from the given temperature, so in this case the heat of the liquid is

$$200 - 32 = 168 \text{ B. t. u.}$$

This result may also be obtained from Plate 3b from the heat of the liquid curve for the temperature of 200° F.

From Plate 1b, for the pressure of 185 pounds and 125° F. superheat we read:

$$\text{Total heat} = 1268 \text{ B. t. u.}$$

Hence the efficiency of boiler and furnace combined is

$$\frac{(1268 - 168)\,9.8}{14300} = 75.5\%$$

20. For each pound of fuel, a certain boiler makes 10 pounds of steam having a quality of 98% and an absolute pressure of 120 pounds per square inch.

If the temperature of the feed water is 80° F., find:

(a) Factor of evaporation.
(b) Equivalent evaporation.

Solution: The heat of the liquid at feed temperature is

$$80 - 32 = 48 \text{ B. t. u.}$$

From Plate 1b, for the pressure of 120 pounds and a quality of 98% we read:

$$\text{Total heat} = 1172 \text{ B. t. u.}$$

Then, by definition, we have:

(a) $$\text{Factor of evaporation} = \frac{\text{Heat absorbed per lb. of steam}}{\text{Latent heat of steam at } 212^\circ \text{ F.}}$$

$$= \frac{1172 - 48}{970.4} = 1.159$$

(b) Equivalent evaporation is the amount of water that would be evaporated from and at 212° F. by the same amount of heat as is actually absorbed per pound of fuel. The equivalent evaporation in this case will therefore be

$$10 \times 1.159 = 11.59 \text{ pounds}$$

21. If, for each pound of fuel, the boiler of the previous problem had delivered 9.5 pounds of steam having a superheat of 175° and at the same pressure as before, find the factor of evaporation and equivalent evaporation.

Solution: From previous problem

$$\text{Heat of the liquid} = 48 \text{ B. t. u.}$$

From Plate 2a, for pressure of 120 pounds and a superheat of 175° we read:

$$\text{Total heat} = 1282 \text{ B. t. u.}$$

Then, as before,

(a) $$\text{Factor of evaporation} = \frac{1282 - 48}{970.4} = 1.273$$

(b) $$\text{Equivalent evaporation} = 9.5 \times 1.273 = 12.1 \text{ pounds.}$$

22. A steam main is supplied with steam from two boilers, one of which furnishes 6,000 pounds of steam per hour, the quality being 98%, while the other boiler furnishes 4,000 pounds of steam per hour, the superheat being 95° F. Assuming that both boilers deliver the steam at a pressure of 190 pounds per square inch

absolute, and neglecting all losses, find the condition of the steam in the main.

Solution: From Plate 1b, for the pressure of 190 pounds and the quality of 98% we read:

Total heat = 1180 B. t. u.

From the same Plate, for same pressure and for a superheat of 95° we read:

Total heat = 1253 B. t. u.

Then for the mixture, we have:

$$\text{Total heat} = \frac{1180 \times 6000 + 1253 \times 4000}{6000 + 4000} = 1209 \text{ B. t. u.}$$

Hence, for the pressure of 190 pounds and a total heat of 1209, from Plate 1b we read:

Superheat in main = 18° F.

23. Assuming no loss by radiation, conduction, or leakage, what would have been the condition of the steam in the steam main of the previous problem, had there been a loss of pressure due to friction in the pipes, so that the pressure in the main was only 175 pounds per square inch absolute?

Solution: Since there is no loss of heat, the total heat of the mixture must be the same as in the previous problem. From Plate 1b, we may follow the 1209-total-heat line until it intersects the 175-pound line, where we read:

Superheat in the main = 20° F.

24. Two boilers, A and B, are connected to the same steam main and the following observations were made:

Total steam passing through the main	=	12000 lbs. per hr.
Average pressure in the main	=	195 lbs. per sq. in. abs.
Average pressure leaving boiler A	=	200 lbs. per sq. in. abs.
Average pressure leaving boiler B	=	210 lbs. per sq. in. abs.
Average superheat in the main	=	56° F.
Average superheat leaving boiler A	=	33° F.
Average superheat leaving boiler B	=	84° F.

Find the weight of steam coming from each boiler, assuming no losses by radiation or conduction.

Solution: From Plate 1b, we read:

For steam from boiler A, having the pressure of 200 pounds and superheat of 33°,

$$\text{Total heat} = 1220 \text{ B. t. u.}$$

For steam from boiler B, having the pressure of 210 pounds and superheat of 84°,

$$\text{Total heat} = 1250 \text{ B. t. u.}$$

For steam in the main, having the pressure of 195 pounds and superheat of 56°,

$$\text{Total heat} = 1232.5 \text{ B. t. u.}$$

Let W_A = lbs. of steam per hr. coming from boiler A
Let W_B = lbs. of steam per hr. coming from boiler B
Let H_A = the total heat of steam B. t. u. per lb. from boiler A
Let H_B = the total heat of steam B. t. u. per lb. from boiler B
Let H_m = the total heat of steam B. t. u. per lb. in main.

Since there has been no loss of heat, we may write:

$$W_A H_A + W_B H_B = (W_A + W_B) H_m$$

From the conditions of the problem

$$W_B = 12000 - W_A$$

Combining these two equations we have:

$$W_A H_A + [12000 - W_A] H_B = 12000 H_m$$

$$\therefore [H_A - H_B] W_A = 12000 [H_m - H_B]$$

$$\therefore W_A = 12000 \left[\frac{H_m - H_B}{H_A - H_B}\right]$$

$$= 12000 \left[\frac{1232.5 - 1250}{1220 - 1250}\right]$$

$$= 12000 \left[\frac{-17.5}{-30}\right]$$

$$= 7000 \text{ pounds per hr.}$$

$$\text{Hence } W_B = 5000 \text{ pounds per hr.}$$

25. The exhaust steam from an engine is delivered to the heating system of a building. If the pressure in this system is maintained constant at 17 pounds per square inch absolute, and if the quality of the steam exhausted by the engine is 88%, find the amount of heat that would be given up by this system when the engine is using 3,000 pounds of steam per hour. The water in the return pipes is delivered to the boiler room at a temperature of 210° F.

Solution: From Plate 3b, for the pressure of 17 pounds and quality of 88% we read:

$$\text{Total heat} = 1037 \text{ B. t. u.}$$

The heat returned to the boiler room is equal to

$$\text{Heat of the liquid} = 210 - 32 = 172 \text{ B. t. u.}$$

The heat given up by the total weight of exhaust steam therefore is

$$3000\ (1037 - 187) = 2577000 \text{ B. t. u. per hr.}$$

26. A 12,000-horse-power steam turbine required 9.6 pounds of steam per horse-power hour when supplied with steam having a pressure of 175 pounds per square inch absolute and 132 degrees of superheat. If the exhaust pressure were 2 inches of mercury absolute, find

(a) Heat supplied per h.p. hr.
(b) Delivered thermal efficiency.
(c) Cycle efficiency.
(d) Theoretical water rate.
(e) Efficiency ratio.

Solution: Let *abcde*, Fig. 4, represent the cycle upon which the ideal turbine operates.

Heat supplied per pound of steam

$$\begin{aligned} &= \text{area } mabcdn \\ &= \text{Total heat}]_d - \text{Heat of the liquid}]_a \\ &= 1270 - 69 = 1201 \text{ B. t. u.} \end{aligned}$$

This value of total heat at the point d is read directly from Plate 1b for the pressure of 175 pounds and superheat of 132°.

The heat of the liquid for the point a is read from Plate 5b, at the intersection of the heat of the liquid curve and the 2 inches of mercury pressure line.

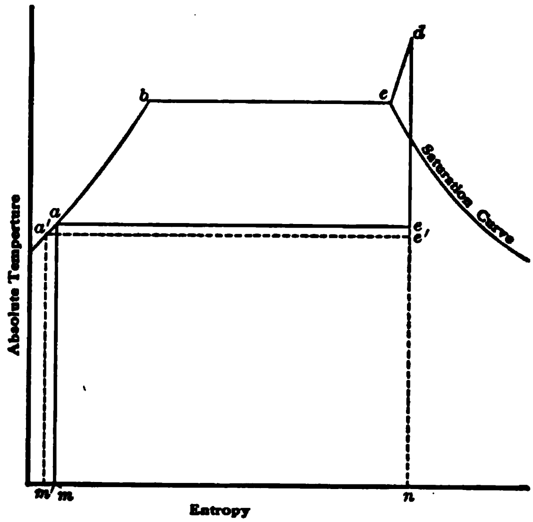


FIG. 4.

Then, since the turbine requires 9.6 pounds of steam per h.p. hour,

(a) Heat supplied per h.p. hr. $= 9.6 \times 1201 = 11529.6$ B.t.u.

(b) Delivered thermal efficiency $= \dfrac{\text{One horse-power hour in B.t.u.}}{\text{Heat supplied per h.p. hr.}}$

$$= \frac{2545}{11529.6} = 22.1\%$$

(c) Cycle efficiency $= \dfrac{\text{Area } abcde}{\text{Area } mabcdn}$

$$= \frac{\text{Total heat}]_d - \text{Total heat}]_e}{1201} = \frac{1270 - 915}{1201}$$

$$= \frac{355}{1201} = 29.55\%$$

This value of total heat for the point e is obtained from Plate 5b, by reading the total heat at the intersection of the pressure line

for 2 inches of mercury and the 1.64 entropy line. This entropy is obtained from Plate 1b, for the state *d*.

(d) Theoretical water rate $= \dfrac{2545}{\text{Net work per lb. of steam}}$

$= \dfrac{2545}{355} = 7.17$ pounds per h.p. hr.

(e) Efficiency ratio $= \dfrac{\text{Actual thermal efficiency}}{\text{Cycle efficiency}}$

$= \dfrac{.221}{.2955} = 74.7\%$

or Efficiency ratio $= \dfrac{\text{Theoretical water rate}}{\text{Actual water rate}}$

$= \dfrac{7.17}{9.6} = 74.7\%$

27. Suppose that all the conditions of operation remain the same as in the previous problem, except the water rate and the vacuum. If, by reducing the exhaust pressure to 1 inch of mercury absolute, the water rate was actually reduced to 9 pounds per h.p. hour, find (a) (b) (c) (d) and (e) as before.

Solution: Referring to the same figure modified by the line $a'e'$ and proceeding in the same manner, we have:

From Plate 5b,

Heat of the liquid$]_{a'} = 47$ B. t. u.

(a) Heat supplied per hr. $= 9\,[1270 - 47] = 11\,007$ B. t. u.

(b) Delivered thermal efficiency $= \dfrac{2545}{11007} = 23.1\%$

From Plate 6b

Total heat$]_{e'} = 882$ B. t. u.

$\therefore$ Net work per cycle $= 1270 - 882 = 388$ B. t. u.

Hence

(c) Cycle efficiency $= \dfrac{388}{1270 - 47} = 31.7\%$

(d) Theoretical water rate $= \dfrac{2545}{388} = 6.56$ pounds per h.p. hr.

(e) Efficiency ratio $= \dfrac{.231}{.317} = 72.8\%$

28. A steam turbine receives steam at a pressure of 80 pounds per square inch absolute and a superheat of 77° F. If this steam expands in a single set of nozzles to a condenser pressure of 1.4 lbs. per square inch absolute, and if we assume no nozzle losses, find, for the steam leaving nozzles:

(a) Quality.
(b) Specific volume.
(c) Velocity.

Solution: From Plate 2b, for the initial state we read:

Total heat = 1223 B. t. u.
Entropy = 1.67

From Plate 5b, for the final pressure and the entropy 1.67, we read:

Total heat = 950 B. t. u.

(a) Quality = 84.4%
(b) Specific volume = 204 cu. ft. per lb.

The energy which has been transformed into velocity is equal to the difference in total heats.

Initial total heat − Final total heat = 1223 − 950 = 273 B. t. u. per lb.

Then, from Table IV, we read:

(c) Velocity = 3696 feet per second.

29. The theoretical steam engine operates on the incomplete expansion cycle, as shown by *abcde*, Figs. 5 and 6. The pressure at the throttle is 165 pounds per square inch absolute and the superheat is 143° F. If the back pressure is 2.8 pounds per square inch absolute and the expansion ratio is 6, find, for the ideal cycle:

(a) Pressure at release.
(b) Quality at release.
(c) Net work of cycle in B. t. u.
(d) Cycle efficiency.
(e) Water rate of an ideal engine working on this cycle.

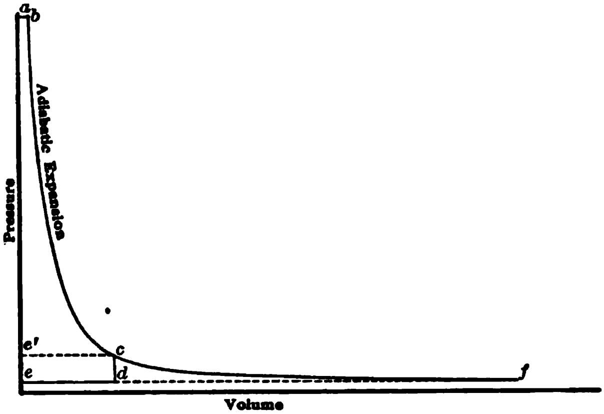


FIG. 5.

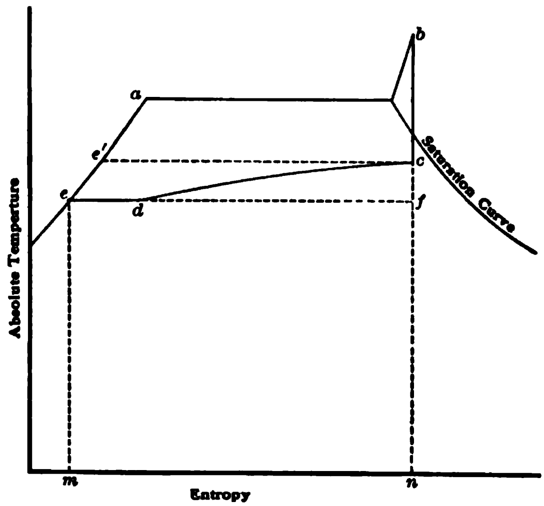


FIG. 6.

Solution: From Plate 1b, for the pressure of 165 pounds and the superheat of 143° we read:

Initial total heat	= 1274 B. t. u. per lb.
Initial specific volume	= 3.4 cu. ft. per lb.
Initial entropy	= 1.65

Since the expansion ratio is 6, the specific volume at release must be

$$V_c = 6\ V_b = 6 \times 3.4 = 20.4 \text{ cu. ft. per lb.}$$

For the ideal cycle, the expansion line is an adiabatic. Hence by following the 1.65 entropy line until it intersects the 20.4 specific-volume line, we may read from Plate 3b:

(a) Release pressure = 18.4 lbs. per sq. in. abs.
(b) Release quality = 93.7%
Total heat$]_c$ = 1094.5 B. t. u.

In order to find the net work of the cycle we may proceed in several ways, one being to subtract the heat rejected from the heat supplied. Referring to the temperature entropy diagram, Fig. 6, we have:

$$\begin{aligned}\text{Heat supplied} &= \text{area } meabn \\ &= \text{Total heat}]_b - \text{Heat of the liquid}]_e \\ &= 1274 - 107 = 1167 \text{ B. t. u. per lb.}\end{aligned}$$

The heat of the liquid at the point e is obtained from Plate 4b for the 2.8 pounds back-pressure line.

$$\begin{aligned}\text{Heat rejected} &= \text{area } cdemn. \\ &= \left.\begin{matrix}\text{Intrinsic}\\ \text{heat}\end{matrix}\right]_c - \left.\begin{matrix}\text{Intrinsic}\\ \text{heat}\end{matrix}\right]_e + \left.\begin{matrix}\text{Work}\\ \text{done}\end{matrix}\right]_c^e \\ &= [1094.5 - 69] - 107 + 11 = 929.5 \text{ B. t. u. per lb.}\end{aligned}$$

This intrinsic heat at c is obtained by subtracting from the total heat for this point the constant pressure external work as obtained from Plate 8b. We know the specific volume at c to be 20.4 cubic feet per pound, and the quality at c has already been found to be 93.7%. Therefore we may read from this plate the value of this external work, 69 B. t. u., as above given.

The intrinsic heat at e is merely the heat of the liquid for the exhaust pressure and has already been found to be 107 B. t. u.

The work done in going from c to e is the same as that done in going from d to e since the work done from e to d is zero, the volume being constant. But the work from d to e may be obtained directly from Plate 8b. Thus by running down the constant specific-volume

line, 20.4, until it intersects the 2.8 pound back-pressure line, we may read:

$$\left.\text{Work done}\right]_d^e = 11 \text{ B. t. u. per lb.}$$

Then from the above values, we have:

$$\text{(c) Net work of cycle} = \text{Heat supplied} - \text{Heat rejected}$$
$$= 1167 - 929.5 = 237.5 \text{ B. t. u. per lb.}$$

$$\text{(d) Cycle efficiency} = \frac{\text{Net work of cycle}}{\text{Heat supplied}} = \frac{237.5}{1167} = 20.35\%$$

$$\text{(e) Ideal water rate} = \frac{2545}{\text{Net work per lb. of steam}} = \frac{2545}{237.5}$$
$$= 10.71 \text{ lbs. per h.p. hr.}$$

NOTE.—For the cycle *abcde*, Fig. 5 would commonly be drawn with a different ratio of scales so that the distance representing the admission pressure would be nearly the same as the distance representing the volume at *d*. In this case, however, it was desired to draw to scale the entire diagram, including the "toe" *cfd*, so that is why the volume at *d* may seem to be entirely too small for any real engine having a ratio of expansion equal to six.

Since a pound of water occupies a space of only about .016 cubic feet, it is evidently impossible in Fig. 5 to represent such a volume with any line other than the zero volume line *ea*. Even with a scale of volumes as large as that used for Fig. 7 with Problem 34, the volume of the water can scarcely be shown.

30. Find the net work of cycle in the previous problem by the method of combining the two cycles *abce′* and *e′cde*, Figs. 5 and 6.

Solution:

$$\left.\begin{array}{l}\text{Net work of}\\ \text{cycle}\\ abce'\end{array}\right] = \left.\begin{array}{c}\text{Total}\\ \text{heat}\end{array}\right]_b - \left.\begin{array}{c}\text{Total}\\ \text{heat}\end{array}\right]_c$$
$$= 1274 - 1094.5$$
$$= 179.5 \text{ B. t. u. per lb.}$$

These values are obtained from Plates 1b and 4b as before.

$$\left.\begin{array}{l}\text{Net work of}\\ \text{cycle}\\ e'cde\end{array}\right] = \left.\text{Work}\right]_e^c - \left.\text{Work}\right]_d^e$$
$$= 69 - 11 = 58 \text{ B. t. u. per lb.}$$

These values may be obtained from Plate 8b as before.

Hence the net work of the cycle *abcde* is

$$179.5 + 58 = 237.5 \text{ B. t. u. per 1b.}$$

31. For the previous problem find the amount of work lost due to incomplete expansion and thereby check the result of preceding problem.

Solution: The work lost due to incomplete expansion is represented by *cfd* in Fig. 5 or 6.

By inspection of these figures it will be seen that

$$cfd = e'cfe - e'cde$$

$$\text{or}\quad \begin{matrix}\text{Work}\\ \text{lost}\end{matrix} = \left\{ \begin{matrix}\text{Total}\\ \text{heat}\end{matrix}\Big]_c - \begin{matrix}\text{Total}\\ \text{heat}\end{matrix}\Big]_f \right\} - \left\{ \text{Work}\Big]_{e'}^{c} - \text{Work}\Big]_d^{e} \right\}$$

$$= 1094.5 - 977.3 - 69 + 11$$

$$= 59.2 \text{ B. t. u. per lb.}$$

The total heat at c is obtained from Plate 3b, as before. The total heat at f is found from Plate 4b, for the given back pressure of 2.8 pounds and the 1.65 entropy line.

$$\text{Work}\Big]_e^c \text{ is found from Plate 8b as before.}$$

$$\text{Work}\Big]_d^{e'} \text{ is found from Plate 8b as before.}$$

To check the net work of the incomplete expansion cycle we may subtract the "toe" from the complete expansion cycle. Thus $abcde = abfe - cfd$

$$\text{But the cycle } abfe = \begin{matrix}\text{Total}\\ \text{heat}\end{matrix}\Big]_b - \begin{matrix}\text{Total}\\ \text{heat}\end{matrix}\Big]_f$$

$$= 1274 - 977.3$$

$$= 296.7 \text{ B. t. u. per lb.}$$

Hence the cycle *abcde* is equal to

$$296.7 - 59.2 = 237.5 \text{ B. t. u. per lb.}$$

This checks the results of 29 and 30.

32. Find the ratio of volumes of the cylinders necessary for the complete and incomplete expansion cycles of problem 29, and also find the per cent. reduction in the net work due to the incomplete expansion.

Solution: From problem 31 we have:

Per cent. reduction in net work is

$$\frac{59.2}{296.7} = 19.9\%$$

From problem 29 the specific volume at e was 20.4 cubic feet per pound.

From Plate 4b, for the given back pressure of 2.8 pounds and the entropy 1.65, we obtain

Specific volume at f = 108.2 cu. ft. per lb.

Then, since the cylinder volumes for any case would be proportional to the specific volumes as thus found for the ideal cycle, we have.

$$\frac{\text{Volume for complete expansion}}{\text{Volume for incomplete expansion}} = \frac{108.2}{20.4} = 5.3$$

33. Find the per cent. increase in the net work of the cycle of problem 30 for a back pressure of 1 pound per square inch absolute instead of 2.8 pounds.

Solution: By referring to the solution of problem 30, we may see that by changing the back pressure only, the solution remains the same except the value of the work done during exhaust. From Plate 8b, running down the specific volume line 20.4, which is the specific volume at release, until we intersect the 1 pound back-pressure line, we obtain

$$\text{Work}\Big]_d^e = 3.8 \text{ B. t. u. per lb.}$$

In problem 30 the work done against the exhaust pressure was 11 B. t. u. per pound. This is an increase of 7.2 B. t. u. per pound.

Hence

Per cent. increase in the net work of the cycle is

$$\frac{7.2}{237.5} = 3.03\%$$

NOTE.—In an actual engine operating under these conditions much less gain than this would be realized on account of greater loss due to cylinder condensation, more work required for vacuum pump, and more heat required to heat the feed water.

34. A direct-acting steam pump operates on a non-expansive cycle as shown by *abcd*, Fig. 7.

The engine is supplied with steam having a pressure of 120 pounds per square inch absolute and a quality of 97%. If the exhaust pressure is 18 pounds per square inch absolute, find the net work of the cycle, and the ideal water rate.

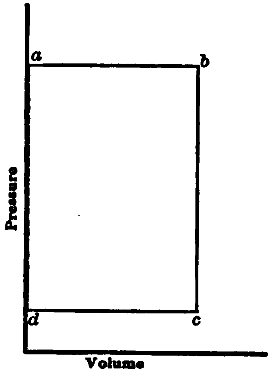


FIG. 7.

Solution: From Plate 8a, for the initial pressure of 120 pounds and quality of 97% we find:

$$\text{Work}\Big]_a^b = 80 \text{ B. t. u. per lb.}$$

Specific volume at b = 3.6 cu. ft. per lb.

Then, running down this constant volume line until we intersect the back pressure of 18 pounds, we have:

$$\text{Work}\Big]_c^d = 12 \text{ B. t. u. per lb.}$$

Then the net work of the cycle becomes

$$80 - 12 = 68 \text{ B. t. u. per lb.}$$

and the ideal water rate is therefore

$$\frac{2545}{68} = 37.4 \text{ lbs. per h.p. hr.}$$

35. In a uni-flow locomotive the clearance is 16% and compression takes place during 90% of the stroke.

If the back pressure is 16 pounds per square inch absolute and the quality of the steam at the beginning of compression is 96%, find the condition of the steam at the end of the stroke, assuming the compression to be adiabatic.

Solution: From Plate 3b, for the pressure of 16 pounds and quality of 96% we may read:

Entropy = 1.6925
Initial specific volume = 23.8 cu. ft. per lb.

We must next find the specific volume at the end of compression, which is equal to the specific volume at the beginning of compression divided by the ratio of compression.

$$\text{Ratio of compression} = \frac{.90 + .16}{.16} = 6.63$$

Hence

$$\text{Final specific volume} = \frac{23.8}{6.63} = 3.59 \text{ cu. ft. per lb.}$$

Then by going along the entropy line 1.6925, until we intersect this specific volume line 3.59, we read from Plate 1a:

Final pressure = 175 lbs. per sq. in. abs.
Final superheat = 240° F.

36. An engine having a steam jacket has a clearance of 15% of the piston displacement which is 3.17 cubic feet. It is found from test results that the weight of steam inside the cylinder during expansion is 0.163 pounds. If release occurs at 95% of the stroke at a pressure of 19 pounds per square inch absolute, find the condition of the steam at release.

Solution: Actual volume of the steam in the cylinder is equal to the clearance volume plus the piston displacement. But since the clearance is given in terms of the piston displacement, we have:

$$\text{Actual volume} = 1.10 \times 3.17 = 3.49 \text{ cu. ft.}$$

Specific volume of the steam in the cylinder is

$$\frac{\text{Actual volume}}{\text{Weight}} = \frac{3.49}{0.163} = 21.4 \text{ cu. ft. per lb.}$$

Then from Plate 3a, for the pressure of 19 pounds and a specific volume of 21.4 we find:

$$\text{Superheat} = 9° \text{ F.}$$

37. The temperature in a condenser is 87° F. and the absolute pressure is equal to 1.43 inches of mercury. Find the weight of air

which must be removed per pound of exhaust steam if the steam entering the condenser has a quality of 85%.

Solution: From the curve giving temperature of vaporization, Plate 5b, for 87°, we find:

$$\text{Pressure of wet steam} = 1.29'' \text{ Hg.}$$

and for this pressure and a quality of 85%

$$\text{Specific volume} = 442. \text{ cu. ft. per lb.}$$

The air pressure is equal to the total pressure less the pressure of the steam, that is,

$$1.43 - 1.29 = 0.14'' \text{ Hg.}$$

1 inch of mercury at 58.1° being equal to 0.49 pound per square inch, the air pressure in the condenser is

$$P = .14 \times .49 \times 144 \text{ lb. per sq. ft.}$$

The absolute temperature in the condenser is

$$T = 460 + 87 = 547°.$$

Hence the weight of air existing in the same space as each pound of steam is

$$w = \frac{PV}{RT} = \frac{(.14 \times .49 \times 144)\, 442}{53.3 \times 547} = 0.15 \text{ lb.}$$

38. If the temperature of the atmosphere is 68.5° F. and the relative humidity is 70%, find the pressure due to this moisture and the weight of moisture in each 1,000 cubic feet of the atmosphere.

Solution: From the temperature of vaporization curve of Plate 6b, for the temperature of 68.5° we may read:

$$\text{Saturation pressure} = 0.7'' \text{ Hg.}$$

Since the relative humidity is almost exactly equal to the

$\dfrac{\text{Actual vapor pressure}}{\text{Saturation pressure}}$, it follows that

$$\text{Actual vapor pressure} = .70 \times 0.7 = 0.49'' \text{ Hg.}$$
$$= .49 \times .49 = 0.24 \text{ lb. per sq. in.}$$

Then from Plate 6a, for this pressure, 0.24 pound and the temperature 68.5° given by the temperature scale on the right-hand side of the sheet, we may read:

Specific volume = 1,310 cu. ft. per lb.
Superheat = 10° F.

It is interesting to observe that this means the vapor in the air in this condition is merely superheated steam having a pressure of 0.24 pound per square inch absolute or 0.49 inch of mercury and a superheat of 10°. Hence the weight of this steam contained in 1,000 cubic feet of space would be

$$\frac{1000}{1310} = 0.764 \text{ lb.}$$

Then by Dalton's law of partial pressures, for the given vapor pressure and temperature, this same weight will be in this space when it is also full of air.

NOTE.—This scale of approximate temperatures, placed on the upper right-hand corner of Plate 6a, is not intended to be sufficiently accurate to solve all hygrometric problems. See page 15 for the variation of this scale from the true temperatures.

39. Find the diameter of pipe to furnish steam for a 9,000 kw. turbine, if it requires 12.8 pounds per kw.-hr. when running at full load with a pressure of 200 pounds per square inch absolute and 145° superheat at the throttle. Allow a velocity of 8,000 feet per minute.

Solution: From Plate 1b, for the steam at the throttle with a pressure of 200 pounds and superheat of 145° we find:

Specific volume = 2.84 cu. ft. per lb.

The area of the pipe in square feet is equal to

$$\frac{\text{Volume in cu. ft. per min.}}{\text{Velocity in ft. per min.}} = \frac{\text{lbs. per kw.-hr.} \times \text{kw.} \times \text{sp. vol.}}{60 \times \text{velocity}}$$

But if the diameter of the pipe in inches is d, then the area in square feet is also equal to $\frac{\pi d^2}{4 \times 144}$.

$$\text{Hence } d = \sqrt{\frac{4 \times 144 \times \text{water rate} \times \text{load} \times \text{sp. vol.}}{\pi \times 60 \times \text{velocity}}}$$

$$= 1.748 \sqrt{\frac{\text{water rate} \times \text{load} \times \text{sp. vol.}}{\text{velocity in ft. per min.}}}$$

$$= 1.748 \sqrt{\frac{12.8 \times 9{,}000 \times 2.84}{8{,}000}}$$

$$= 11.16''.$$

This means, of course, that a 11-inch pipe would be used.

40. The Parson's turbine at the Fiske Street Station of the Commonwealth Edison Co., Chicago, has an exhaust opening to the condenser of 252 square feet. If the water rate for a back pressure of 1 inch of mercury and a load of 25,000 kw. is 11.65 pounds per kw.-hr., find the velocity through this opening, assuming the steam has a quality of 80%.

Solution: From Plate 6b, for the pressure of 1 inch of mercury and quality of 80% we find:
Specific volume of exhaust steam = 523 cu. ft. per lb.
Then

$$\text{Velocity} = \frac{\text{Volume}}{\text{Area}} = \frac{11.65 \times 25{,}000 \times 523}{60 \times 252} = 10{,}080 \text{ ft. per min.}$$

41. If, in the second stage of a Curtis turbine, the pressure at the entrance to the nozzle is 56 pounds per square inch absolute, the superheat 64°, and the pressure in the next stage 36.8 pounds per square inch absolute, find the cross-sectional area of each nozzle for this stage in order that 112 such nozzles will give a flow of 96,000 pounds of steam per hour. Assume the coefficient of velocity for this condition to be 96%.

Solution: From Plate 2b, for a pressure of 56 pounds and superheat of 64° we find:

Total heat = 1,208 B. t. u.
Entropy = 1.69

Then, by following this entropy line until we intersect the 36.8-pound line on Plate 3a, we may read:

Total heat = 1,173 B. t. u.
Specific volume = 11.6 cu. ft. per lb.

Then the theoretically available energy to produce velocity in this stage is

$$1{,}208 - 1{,}173 = 35 \text{ B. t. u. per lb.}$$

and from Table IV this velocity is 1,323 feet per second.

For the given nozzle coefficient the velocity would therefore be

$$.96 \times 1{,}323 = 1{,}270 \text{ ft. per sec.}$$

Let A = area in sq. in. of each nozzle.
Let N = number of nozzles.
Let v = velocity in ft. per sec.
Let V = specific volume of steam as it passes the throat.

Then the flow in pounds per hour is

$$F = \frac{3{,}600\,N\,A\,v}{144V} = \frac{25\,N\,A\,v}{V}$$

$$\therefore A = \frac{F\,V}{25\,N\,v} = \frac{96{,}000 \times 11.6}{25 \times 112 \times 1270} = 0.313 \text{ sq. in.}$$

For this and the following problem it will be observed that the ratio of pressures between stages is considerably over 57%, which means that the maximum velocity occurs at the throat of the nozzle and that the nozzle will therefore have no divergent part.

42. Find the size of nozzles and the pressures in the third and fourth stages for the turbine of the previous problem if in each of these stages there are to be 82 nozzles and the theoretically available energy is to be 35 B. t. u. per stage. Assume that the nozzle, bucket, and rotational losses amount to 25% of this energy. Neglect

radiation loss and leakage of steam and assume nozzle coefficient to be 96% as before.

Solution: The losses in each stage go to reheat the steam, and this reheating is assumed to take place at constant pressure after the expansion in the nozzle, since the nozzle loss is very small. Hence, from the previous problem, the steam enters the third-stage nozzle in the following condition:

Pressure = 36.8 lbs.
Total heat = $1{,}173 + .25 \times 35 = 1{,}181.8$ B. t. u.

Then, from Plate 3a, for these two values we obtain:

Entropy = 1.70+
Superheat = 27°

Since it is desired to give up 35 B. t. u. in this stage, we now follow down this entropy line until the total heat becomes

$1{,}181.8 - 35 = 1{,}146.8$ B. t. u.

and from Plate 3b we find for this total heat and entropy of 1.70+

Pressure = 23.5 lbs.
Specific volume = 17.1 cu. ft.

The velocity corresponding to 35 B. t. u. is from Table IV equal to 1,323 feet per second. Hence, using the same notation as before, the area of the throat of the nozzle for the third stage is

$$A_3 = \frac{F\,V}{25\,N\,v} = \frac{96{,}000 \times 17.1}{25 \times 82 \times (.96 \times 1{,}323)} = 0.632 \text{ sq. in.}$$

Then the steam enters the nozzle of the fourth stage in the following condition:

Pressure = 23.5 lbs.
Total heat = $1{,}146.8 + .25 \times 35 = 1{,}155.6$

From these two values and Plate 3a we obtain

Entropy = 1.713
Quality = 99.5%

and by following this entropy line until the total heat becomes

$$1{,}155.6 - 35 = 1{,}120.6 \text{ B. t. u.}$$

we have from Plate 3b

$$\begin{aligned} \text{Pressure} &= 14.6 \text{ lbs.} \\ \text{Specific volume} &= 26.2 \text{ cu. ft.} \end{aligned}$$

Hence the area of the throat of the nozzle for the fourth stage is

$$A_4 = \frac{FV}{25\ Nv} = \frac{96{,}000 \times 26.2}{25 \times 82 \times (.96 \times 1{,}323)} = 0.967 \text{ sq. in.}$$

43. Steam expands through a properly shaped divergent nozzle from an initial pressure of 125 pounds per square inch absolute and 220° of superheat to a final pressure of 1 pound per square inch absolute. Find the proper cross-sectional areas of this nozzle to permit a flow of one pound per second, assuming adiabatic expansion and no friction.

Solution: From Plate 2a, for the pressure of 125 pounds and superheat of 220° we may read:

$$\begin{aligned} \text{Total heat} &= 1{,}305 \text{ B. t. u.} \\ \text{Entropy} &= 1.71 \end{aligned}$$

and from Plate 5b for this entropy and a pressure of 1 pound we find:

$$\begin{aligned} \text{Total heat} &= 955 \text{ B. t. u.} \\ \text{Specific volume} &= 285 \text{ cu. ft.} \end{aligned}$$

Then the total available energy is

$$1{,}305 - 955 = 350 \text{ B. t. u. per lb.}$$

Let the nozzle be divided into sections, so that in each one the expansion will be sufficient to use equal parts of this total available energy. If we compute the velocity for 10 such sections we shall have 35 B. t. u. liberated in each one. By the charts we may now follow the entropy line 1.71 until it intersects the desired total heat line, and then read the corresponding pressure and specific

volume. By Table IV the velocity is obtained, and the area of the nozzle at any section may then be found from the equation

$$\text{Area in sq. in.} = \frac{(\text{Cu. ft. flowing per sec.})144}{\text{Velocity in ft. per sec.}}.$$

Hence the following values may be at once tabulated:

Section	Entropy	Available Energy B. t. u. per Lb.	Total Heat B. t. u.	Pressure Lbs. per Sq. In. Abs.	Specific Volume Cu. Ft. per Lb.	Velocity Ft. per Second	Area of Nozzle Sq. In.	Plate Used
Entrance	1.71	0	1,305	125.	4.82	....	.....	2a
1	1.71	35	1,270	89.8	6.18	1,323	0.674	2a
2	1.71	70	1,235	62.8	8.10	1,872	0.624	2a
3	1.71	105	1,200	42.4	10.9	2,292	0.685	2b
4	1.71	140	1,165	27.6	15.1	2,647	0.822	3a
5	1.71	175	1,130	17.1	22.7	2,960	1.101	3b
6	1.71	210	1,095	10.3	35.4	3,241	1.572	3b
7	1.71	245	1,060	6.10	56.5	3,501	2.32	4b
8	1.71	280	1,025	3.44	94.0	3,743	3.62	4b
9	1.71	315	990	1.90	160.	3,970	5.81	5b
10	1.71	350	955	1.00	285.	4,185	9.82	5b

If these sections are now laid off at equal intervals so that the total length becomes equal to the desired length of the nozzle the form will be such that it will give uniform acceleration of the steam. This is theoretically desirable, and is due to the fact that the sections have been chosen at such points that the energy transformed into velocity in each one is constant. Such a nozzle will have curved elements for its divergent part and will therefore be expensive to make. It has been found by experiment that the divergent part of the nozzle can be made of straight-line elements without any great loss in efficiency. Since there is a large amount of hand work required in making nozzles, they are therefore nearly always made with the divergent part as the frustum of a pyramid or cone. The throat of the nozzle must be approached by a gradually decreasing cross-sectional area in order to prevent loss due to the sudden convergence of the stream lines.

No general rule can be given to determine the best length of nozzle for all conditions. The length of the rounded entrance to the

throat is a very small portion of the total length of the nozzle, and it is common practice for some designers to have the divergent part of the nozzle taper about one in twelve, while others make it only about one in twenty.

44. For the expansion of the previous problem find the value of the exponent n which satisfies the equation $p_0 v_0^n = p\, v^n$ where p represents the pressure and v the specific volume of the steam at any time during its passage through the nozzle and p_0 and v_0 refer to the pressure and specific volume at the entrance to nozzle.

Solution: Rewriting the equation we have

$$\frac{p_0}{p} = \left(\frac{v}{v_0}\right)^n$$

Then by the aid of a log log slide rule, or logarithms, and the results of the previous problem the following table may at once be made:

Section	p	v	$\frac{p_0}{p}$	$\frac{v}{v_0}$	n	Quality or Superheat	
						Value	Plate Used
0	125.	4.82	1.000	1.000	....	220°	2a
1	89.8	6.18	1.392	1.283	1 325	166°	2a
2	62.8	8.10	1.990	1.680	1.325	112°	2a
3	42.4	10.9	2.950	2.265	1.325	60°	2b
4	27.6	15.1	4.535	3.131	1.325	5°	3a
5	17.1	22.7	7.32	4.71	1.285	97.6%	3b
6	10.3	35.4	12.14	7.35	1.250	95.1%	3b
7	6.10	56.5	20.50	11.73	1.227	92.6%	4b
8	3.44	94.0	36.38	19.50	1.210	90.2%	4b
9	1.90	160.	65.80	33.2	1.195	87.9%	5b
10	1.00	285.	125.00	59.1	1.185	85.6%	5b

This table shows that so long as the steam was superheated the value of n was constant, but decreased rapidly after reaching the wet region. From Plate 3a, it may be seen that the 1.71 entropy line crosses the saturation curve at a pressure of a little more than 26 pounds.

45. With the value of $n = 1.325$ as obtained from problem 44, find the pressure at the throat of this nozzle by the following equation:*

$$\frac{p_t}{p_0} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \text{ where}$$

p_t = pressure at the throat of the nozzle.
p_0 = pressure at entrance to nozzle.

Solution:

$$\begin{aligned} p_t &= p_0 \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \\ &= 125 \left(\frac{2}{2.325}\right)^{\frac{1.325}{0.325}} \\ &= 125 \left(\frac{1}{1.1625}\right)^{4.075} \\ &= 125 \left(\frac{1}{1.848}\right) = 67.75 \text{ lbs.} \\ &= 54.2\% \text{ of } p_0. \end{aligned}$$

46. With the throat pressure, as determined from the equation given in the previous problem, determine the area of the throat and then find by trial whether this is the minimum section.

Solution: Taking pressures each side of the throat pressure as found above, and obtaining the values needed as in problem 43, we may construct the following table:

Section	Entropy	Pressure	Total Heat B. t. u.	Available Energy B. t. u.	Velocity Ft. per Sec.	Specific Volume	Area Sq. In.
0	1.71	125.	1,305.	0	....	....	....
a	1.71	70.	1,245.8	59.2	1,721	7.46	0.624
b	1.71	68.	1,242.9	62.1	1,762	7.62	0.623
Throat	1.71	67.75	1,242.5	62.5	1,767	7.64	0.622
c	1.71	67.	1,241.5	63.5	1,782	7.70	0.622
d	1.71	66.	1,240.	65.	1,803	7.80	0.623

* For the derivation of this equation see any good book on turbine nozzles.

From the above results, it is seen that the value of the throat pressure, as found from the equation of problem 45, is checked as well as can be desired. The computations from the charts and slide rule indicate that the pressure of 67 pounds would answer equally well.

47. Supposing that the steam is compressed, adiabatically, from the final condition given in problem 43, to the pressure of 125 pounds, find the value of *m* to satisfy the equation $p_{10} v_{10}^m = pv^m$
The subscript 10 refers to the state of the steam when in the section 10 of problem 43.

Solution: The equation may be put in the more convenient form

$$\frac{p}{p_{10}} = \left(\frac{v_{10}}{v}\right)^m$$

Section	p	v	$\frac{p}{p_{10}}$	$\frac{v_{10}}{v}$	m	Quality or Superheat from 44
10	1.00	285.	1.	1.	.....	85.6%
9	1.90	160.	1.9	1.783	1.110	87.9%
8	3.44	94.	3.44	3.03	1.115	90.2%
7	6.10	56.5	6.10	5.05	1.117	92.6%
6	10.3	35.4	10.3	8.05	1.118	95. %
5	17.1	22.7	17.1	12.55	1.122	96.6%
4	27.6	15.1	27.6	18.88	1.128	5°
3	42.4	10.9	42.4	26.12	1.148	60°
2	62.8	8.10	62.8	35.2	1.163	112°
1	89.8	6.18	89.8	46.2	1.175	166°
0	125.	4.82	125.	59.1	1.185	220°

From an inspection of these exponents and those as given by the results of 44, it will be seen that the value of the exponent for adiabatic compression or expansion depends upon the state of the steam and a simple general expression to determine its value cannot be given.

When any adiabatic expansion or compression of steam is plotted on the P-V diagram the resultant curve will always be smooth, even though the variation in the exponent may be considerable as in the above cases. This means that the exponents change gradually from one to the other even when crossing the saturation curve. The reason that the above tabulations do not show this more fully is due to the fact that the desire to make a short table necessitated the selection of points which are some distance apart.

48. In making a turbine test, the barometer was read as 29.55 inches at a temperature of 50° F. The diameter of the barometer tube was 0.25 inch and the height of the meniscus was 0.02 inch. The vacuum was determined by reading the height of mercury in a glass tube 0.2 inch in diameter, the bottom end of which was resting in a vessel of mercury. The height of this mercury column was 28.85 inches at a temperature of 77° F. and the meniscus was 0.04 inch. The elevation of the condenser was 500 feet above sea level and the barometer was in another building 50 feet higher than this. The temperature of the atmosphere was 40° F.

Find the pressure in the condenser in inches of mercury.

Solution: The barometer corrections are:

(1) Due to capillarity from Table II	+ .012 in.
(2) Due to temperature from Plate 9a	+ .024 "
(3) Due to change in elevation from Plate 9b	+ .055 "
Total	+ .091 in.

This third correction is obtained by running along the 40° temperature line until we intersect the line representing the average altitude, which for this case would be 525 feet, where the correction for 100 feet is seen to be .110. Then for 50-foot change in elevation the correction would be one-half of this, or .055, as above.

The corrections to the mercury column attached to condenser are:

(1) Due to capillarity from Table II	+ .045 in.
(2) Due to temperature from Plate 9a	− .053 "
Total	− .008 in.

Hence by using correction to nearest hundredth
The barometer is

$$29.55 + .09 = 29.64 \text{ in. at } 58.1^\circ$$

and the vacuum is

$$28.85 - .01 = 28.84 \text{ in. at } 58.1^\circ$$

and absolute pressure in condenser is

$$29.64 - 28.84 = 0.8 \text{ in. Hg.}$$

NOTE.—Without making any of the above corrections the pressure in the condenser would appear to be 29.55 — 28.85 = 0.7 in. Hg. and from Plate 6b the difference in total heats between a pressure of 0.7 in. and 0.8 in. for some common entropy line is seen to be about 6 or 7 B. t. u. This amount of heat might easily mean 2% of the total available energy or as much as 15 or 20% of the energy available in the last stage.

49.

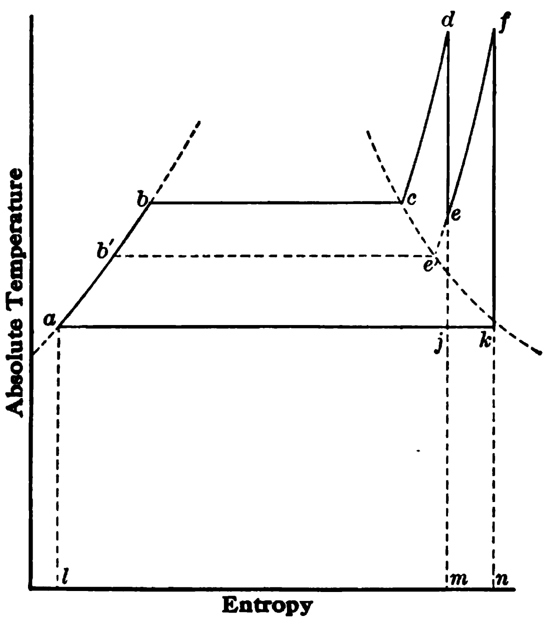


FIG. 8.

Let the area *abcdefka*, Fig. 8, represent the net work in B. t. u. of the theoretical cycle for the Ferranti turbine.*

* See Power, December 30, 1913, page 908, for a description of this turbine.

If the steam is received at a pressure of 145 pounds per square inch absolute and 360 degrees of superheat, then expands, adiabatically, to 25 pounds per square inch absolute, and, after that, is reheated at constant pressure to the initial temperature and then finally expands, adiabatically, to the back pressure of 1.5 inches of mercury, find

(a) Net work of the cycle.
(b) Heat supplied per pound of steam.
(c) Cycle efficiency.
(d) Theoretical water rate.

Solution: From Plate 2a, for the pressure of 145 pounds and superheat of 360° we find

$$\text{Total heat}\Big]_d = H_d = 1{,}377 \text{ B. t. u.}$$

$$\text{Entropy}\Big]_d = 1.76$$

and from Plate 3a for this entropy and a pressure of 25 pounds we have

$$\text{Total heat}\Big]_e = H_e = 1{,}195 \text{ B. t. u.}$$

From Plates 1b and 3b respectively, we also find:
Temperature of vaporization for 145 lbs. $= t_b = 356°$ F.
Temperature of vaporization for 25 lbs. $= t_{b'} = 240°$ F.
Hence

$$t_d = 356 + 360 = 716° \text{ F.}$$

and since the temperature after reheating is to be the same as this, the superheat at f becomes

$$t_f - t_{e'} = 716 - 240 = 476°$$

Then, from Plate 7, for the pressure of 25 pounds and 476° of superheat we have

$$\text{Total heat}\Big]_f = H_f = 1386 \text{ B. t. u.}$$

$$\text{Entropy}\Big]_f = 1.959+ \quad \text{or } 1.96-$$

For this entropy and the condenser pressure of 1.5 inches of mercury, we may then obtain from Plate 5a

$$\text{Total heat}\Big]_k = H_k = 1077$$

and from Plate 5b the heat of the liquid corresponding to the pressure of 1.5 inches of mercury is.

$$h_a = 60 \text{ B. t. u.}$$

Then by inspection of Fig. 8 we have:

(a) Net work of cycle $= \text{area } b'bcdee' + b'e'fka$
$= H_d - H_e + H_f - H_k$
$= 1377 - 1195 + 1386 - 1077$
$= 491$ B. t. u. per lb. of steam.

(b) Heat supplied per lb. of steam $= \text{area } labcdefn$
$= H_d - H_e + H_f - h_a$
$= 1377 - 1195 + 1386 - 60$
$= 1508$ B. t. u.

(c) Cycle efficiency $= \dfrac{491}{1508} = 32.6\%$

(d) Theoretical water rate is $\dfrac{2545}{491} = 5.19$ lbs. per h.p. hr.

or $\dfrac{3412}{491} = 6.94$ lbs. per kw. hr.

50. Supposing the ordinary steam turbine cycle *abcdj*, Fig. 8, had been followed, find (a) (b) (c) (d) as before.

Solution: The only additional numerical value needed is the total heat at the point *j*. From Plate 5b, for the entropy 1.76 and the pressure of 1.5 inches of mercury we find:

$$\text{Total heat}\Big]_j = H_j = 967.3 \text{ B. t. u.}$$

Hence

(a) Net work of cycle $= H_d - H_j = 1377 - 967.3 = 409.7$ B. t. u.

(b) Heat supplied per pound $= H_d - h_a = 1377-60=1317$ B. t. u.

(c) Cycle efficiency $= \dfrac{409.7}{1317} = 31.15\%$

(d) Theoretical water rate $= \dfrac{2545}{409.7} = 6.22$ lbs. per h.p. hr.

$$\text{or} \quad \frac{3412}{409.7} = 8.32 \text{ lbs. per kw. hr.}$$

From the above results it is seen that for the theoretical cycles the decrease in water rate due to the reheating is

$$\frac{8.32 - 6.94}{6.94} = 19.9\%$$

and the increase in heat supplied is

$$\frac{1508 - 1317}{1317} = \frac{191}{1317} = 14.5\%$$

or from the cycle efficiencies the gain due to reheating is

$$\frac{.326 - .3115}{.3115} = 4.66\%$$

In an actual turbine built to operate on this cycle there would be the disadvantages of higher first cost, more complications, and greater radiation loss. On the other hand, by having the steam superheated for the entire passage through the turbine, or nearly so, the leakage, friction, and rotational losses are very much reduced. Whether the reduction of these losses and the slight increase in the cycle efficiency will be sufficient to make this type of turbine superior to others cannot yet be told.

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